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Investigation of the heat recovery system for LB20 and LB21 in Building 99, University of Gävle

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2016

Resumo

Sistemas de aquecimento, ventilação e ar condicionado (AVAC) encontram-se amplamente distribuídos por todo o mundo devido à sua capacidade de ajustamento de alguns parâmetros climáticos locais, como temperatura, humidade relativa, limpeza e distribuição do ar, até níveis desejáveis que são passíveis de verificação num hipotético clima ideal. Um estudo de revisão acerca do uso de energia nos edifícios em países desenvolvidos mostra que, atualmente, esta tecnologia é responsável por uma porção de cerca de 20% do uso final de energia nos edifícios, aumentando até 50% em países cujo clima é quente e húmido, havendo por isso maior necessidade de controlar o ambiente interior de modo a garantir que existe qualidade do ar e conforto térmico para os ocupantes. Visando o decréscimo destes valores de uso de energia, é possível verificar cada vez mais que diferentes sistemas de recuperação de calor inseridos em sistemas AVAC têm vindo a ser desenvolvidos e utilizados ao longo das últimas décadas. Hoje em dia, é obrigatório instalar um desses sistemas quando, por exemplo, o valor projetado do fluxo de ar a ser fornecido, está acima do limite que permite evitar o uso dos mesmos, facto que é possível de verificar sobretudo, em edifícios não residenciais. Devido às diferenças encontradas entre cada instalação de sistemas AVAC em edifícios com diferentes características, existem diversos tipos de unidades de recuperação de calor que podem ser inseridas no sistema, tendo cada uma destas as suas próprias características de performance e funcionamento, tornando-a assim mais ou menos indicada para fazer parte de um certo sistema de ventilação, em particular. A ventilação com recuperação de energia do tipo ar para ar, é baseada na transferência de calor (somente sensível ou também latente) recuperado do fluxo de ar com maior temperatura para o fluxo de ar com temperatura mais baixa, pré-aquecendo (no período de inverno) o ar que vem do exterior e que será fornecido no edifício. Portanto, é importante compreender como é que estas unidades de recuperação de calor funcionam e em que situações têm que ser usadas, não esquecendo a visualização do seu efeito final no sistema de AVAC.

Os principais objetivos deste estudo são a investigação do atual funcionamento das unidades de recuperação de calor nos sistemas LB20 e LB21 que fazem parte do sistema geral de AVAC do edifício 99, na Universidade de Gävle, e fazer algumas sugestões que levassem à melhoria da sua eficiência atual. Além disso, as taxas de transferência de energia sob a forma de calor sensível associadas a estas unidades foram calculadas de modo a perceber o impacto que estas têm no sistema geral de AVAC, assim como a oportunidade financeira advinda de pequenos aperfeiçoamentos nas mesmas. Para avaliar o sistema, os valores momentâneos de alguns parâmetros foram recolhidos na corrente de ar e na solução de etilenoglicol que funciona como meio de transferência de calor entre a corrente de ar que entra e a que sai do sistema, encontrando-se este confinado a circular num circuito de tubos que fazem parte das atuais unidades de recuperação de calor, neste caso do tipo bateria *run-around*. Uma vez que o cálculo da eficiência das unidades de recuperação de calor é dependente dos fluxos de entrada e de saída do ar do sistema AVAC, os fluxos de ar foram recolhidos juntamente com as temperaturas do mesmo. Para visualizar o ponto de funcionamento das bombas hidráulicas em cada um dos sistemas e poder comparar com o ponto desejado, foi também recolhido o fluxo e a diferença de pressão da solução de etilenoglicol. Os valores relativos a transferências de calor no sistema, assim como a atual quantidade de calor sensível recuperado através do uso da unidade de recuperação de calor, a quantidade de calor necessária para reaquecer o ar até à temperatura desejada e o valor teórico máximo de calor possível de transferir, foram calculados tendo em conta os valores de fluxo e temperatura do ar.

Depois dos devidos cálculos, obteve-se que, para o dia mais frio de medições (fevereiro), a eficácia sensível naquele instante de medição era de 42% em LB20 e 47% em LB21, alterando para 44% e 43% no dia mais quente (abril), respetivamente, estando abaixo dos valores típicos esperados para este tipo de recuperador de calor em sistemas AVAC, que variam entre 55 e 65%. O calor que é de facto transferido pela unidade de recuperação de calor e que representa as poupanças da energia extra a aplicar na corrente de ar a ser fornecida para que esta atinja a temperatura desejada, é maior no dia mais frio, com o valor de 46 kW em LB20 e 84 kW em

LB21, justificando assim a existência de tais unidades, mesmo que estas impliquem o uso de bombas hidráulicas para assegurar a circulação do fluido de transferência de calor. Tendo em conta a eficiência estimada para cada bomba hidráulica, assim como o fluxo do fluido circulante, obteve-se um consumo individual de cerca de 1.1kW. Os reduzidos valores de eficiência mostraram que ambas as unidades de recuperação de calor estão a trabalhar abaixo do desejado, sendo que o mesmo se verifica nas bombas hidráulicas que fazem parte das mesmas unidades. Este facto, a par da degradação das unidades que foi possível observar no local devido a fatores como a falta de manutenção, indicam que uma limpeza completa (seguida de uma substituição do atual fluido de transferência de calor (etilenoglicol) para o desejado (propilenoglicol)) das unidades de recuperação de calor e um novo ajuste das bombas e válvulas no caso de mudanças posteriores, são necessários. Ao fazer isto espera-se ver um aumento da eficácia sensível média anual para cerca de 45% em ambas as unidades, o que levará a uma potencial poupança económica, de aproximadamente 41 000 SEK, por ano, para o conjunto das duas unidades.

Palavras-chave:

Sistemas de aquecimento, ventilação e ar condicionado, ventilação com recuperação de calor, bateria *run-around*, rácio de temperatura, eficácia sensível.

Abstract

Heating, ventilation and air-conditioning (HVAC) systems are widely distributed over the world due to their capacity to adjust some local climate parameters, like temperature, relative humidity, cleanliness and distribution of the air until the desired levels verified in a hypothetical ideal climate. A review of buildings' energy usage in developed countries shows that in the present this energy service is responsible for a portion of about 20% of the final energy usage on them, increasing up to 50% in hot-humid countries. In order to decrease these values, more and more different heat recovery systems have been developed and implemented over the last decades. Nowadays it is mandatory to install one of these units when the design conditions are above the limit values to avoid such components, what is possible to verify mostly in non-residential buildings. Each one of those units has its own performance and working characteristics that turns it more indicated to make part of a certain ventilation system in particular. Air-to-air energy recovery ventilation is based on the heat recovery transfer (latent and/or sensible) from the flow at high temperature to the flow at lower temperature, pre-warming the outdoor supply air (in the case of the winter). Therefore, it is important to understand in which concept those units have to be used and more important than that, how they work, helping to visualize their final effect on the HVAC system.

The major aims of this study were to investigate the actual performance of the heat recovery units for LB20 and LB21 in building 99 at the University of Gävle and make some suggestions that could enhance their actual efficiency. Furthermore, the energy transfer rates associated to the heat recovery units were calculated in order to understand the impact of such components in the overall HVAC system as also the possible financial opportunity by making small improvements in the same units. To assess the system, values of temperature and flow (among others) were collected in the air stream and in the ethylene-glycol solution that works as heat transfer medium between air streams and is enclosed in pipes that make part of the actual run-around heat recovery units.

After some calculations, it was obtained that for the coldest day of measurements, the sensible effectiveness was 42% in LB20 and 47% in LB21, changing to 44% and 43% in the warmer day, respectively. The actual heat transfer representing the savings in the supply air stream is higher on the coldest day, with values of 46 kW in LB20 and 84 kW in LB21, justifying the existence of the heat recovery units even if those ones imply the use of hydraulic pumps to ensure the loop. The low values of efficiency have shown that both heat recovery units are working below the desired performance similarly to the pumps that make part of the same units. This fact, together with the degradation of the units that is possible to observe in the local, indicates that a complete cleaning (followed by a change of the heat transfer medium) of the heat recovery units and a new adjustment of pumps and valves for the further changes, are necessary. By doing this, it is expected to see the year average sensible effectiveness increase to close to 45% in both units which will lead to a potential economic saving of around 41 000 SEK per year.

Keywords:

Heating, ventilation and air conditioning systems, heat recovery ventilation, run-around heat recovery, temperature ratio, sensible effectiveness

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Acknowledgments

During this period working on my dissertation in the programme of Energy Systems at University of Gävle several persons were very helpful. They supported me to carry on this project with success until the end, not only answering all my questions and teaching me but also being my lighthouse and giving me tools to construct wisely a foundation for a prosperous future. I am very grateful for the opportunity that was given to me to perform my final master thesis in this university.

First of all I would like to thank to Roland Forsberg for all the patience, enthusiasm, dedication and information that he passed to me over these last 6 months, helping me by all his comments and instructions and by supporting me in all the necessary moments. Moreover, I would like to thank to him for giving me the opportunity to write this study and specially for transmitting me all the excitement and happiness when doing something that we really like.

I would like to thank to Professor Nawzad Mardan, energy systems program director, not only for all the patience since the beginning of the year and all the explanations regarding to my master thesis here as an exchange student, but also for all the indispensable guidance on it.

I would like to convey my sincere gratitude to Magnus Jagbrant from Akademiska Hus, for all the crucial help since the beginning of the project until the end, being very precise and professional without decline a single time that I asked my never-ending questions. All the information provided by him was fundamental to analyse the units.

The biggest thanks of all I dedicate to my family, in special to my parents, my siblings and my grandparents for all the education and freedom that they gave to me over these 23 years, helping me to shape the person that I am today. A special thanks to my loyal friends from my home university and hometown that were always ready to help and support me with their true love and friendship, especially in the hardest times of the elaboration of this thesis.

Last but not least, a devote thank you to all my colleagues in the University of Gävle, not only enrolled in the programme of Energy Systems but also in other studies, for all the guidance and friendship during my period of studies there.

Abbreviations and nomenclature

AHU	Air Handling Unit
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
CAV	Constant Air Volume
ERV	Energy Recovery Ventilation
IAQ	Indoor Air Quality
IECC	International Energy Conservation Code
ISO	International Organization for Standardization
HRV	Heat Recovery Ventilation
HVAC	Heating, Ventilation and Air Conditioning
NTU	Number of Transfer Units
PMV	Predicted Mean Vote (-)
PPD	Predicted Percentage Dissatisfied (-)
SMACNA	Sheet Metal and Air Conditioning Contractors' National Association
VAV	Variable Air Volume
WBGT	Wet Bulb Globe Temperature

C_p	Specific heat, (kJ/(kg.K))
g	Acceleration of the gravity, (m/s^2)
H	Head, (m)
$\dot{V}$	Volumetric flow rate, (m^3/s)
$\dot{V}_e$	Supply flow rate, (m^3/s)
$\dot{V}_{min}$	Minimum value between supply and extract flow rates, (m^3/s)
q_{max}	Theoretical maximum sensible heat transfer rate by the heat recovery unit, (kW)
q_S	Heat transfer needed without heat recovery unit, (kW)
q_{hc}	Extra heat transfer by the reheating coil, (kW)
q_{actual}	Actual heat recovery by the heat recovery unit, (kW)
R	Ratio between supply airflow and minimum airflow between supply and extract, (-)
$Ratio$	Ratio between supply and exhaust airflow (-)

T_C	Air temperature after the reheating coil, (ºC)
T_e	Outdoor air temperature, (ºC)
T_{ext}	Extract air temperature, (ºC)
T_s	Supply air temperature, (ºC)
T_{rec}	Air temperature after the preheating coil, (ºC)
T_{exh}	Exhaust air temperature, (ºC)

Greek Symbols

ε	Sensible effectiveness, (-)
η_p	Hydraulic efficiency of a pump, (-)
η_{Tb}	Temperature ratio for a balanced system, (-)
η_T	Temperature ratio for a unbalanced system, (-)
ρ	Fluid density, (kg/m^3)
ρ_{air}	Density of the air, (kg/m^3)
ΔP	Difference of pressure, (Pa)

Chapter 1 – Introduction

1.1. Background

Nowadays there is a distinct rise in the concern about the environment, focused on the reduction of greenhouse gases and consequently the decrease of energy usage. Despite that, there is also a growing interest on the wellbeing and human comfort directly related with energy usage, working against the previously desired energy savings. Buildings are one of the principal sectors regarding the energy usage, as indicated by the lasts reports from European Commission an energy usage of 40% from them, corresponding to 36% of the total amount of CO_2 emissions in the UE [1].

In order to boost the productivity levels of citizens, in the last years, some facilities that help to control the indoor environment have been installed, turning the indoor life better and healthier. These facilities are used all year round in order to adjust the local time climate conditions to conditions possible to verify in a hypothetical ideal climate. Heating, Ventilation and Air Conditioning (HVAC) systems provide the correct amount of airflow, heat and coolness to each room in a building having also the ability of conditioning the air, changing, for example, its levels of relative humidity. The control of parameters as temperature, humidity and air quality (pollutants) is globally essential but with special attention to certain buildings as schools, hospitals and factories and also in controlled spaces like laboratories. Nowadays, only few commercial buildings and offices do not include this kind of thermal systems. With the current intensive use of HVAC systems in buildings, is known that about 10-20% of final energy usage of a building in developed countries is due to these systems [2], increasing up to 50% in tropical climates [3].

Due to the urgent need of energy savings strongly motivated by financial savings, the planning of ventilation systems is gradually facing bigger challenges once every little detail can lead to a growth of energy efficiency and consequently a decrease of energy usage. A well designed HVAC system would help to reduce the energy usage considerably and spare the environment due to the huge scale of the global topic. The addition of a heat recovery system in such devices is one of the possibilities to reduce the energy usage since it recovers a part of heat from the exhaust air to preheat/precool the supply outdoor air, that otherwise would be wasted to the atmosphere. The actual existence of such components is the proof of a better design and inclusion of innovative technology to enhance the performance of high quality ventilation systems. Hereupon, it is extremely important to have heat recovery systems working at their maximum possible efficiency point. In this text are presented some possibilities of air-to-air energy recovery devices, with special emphasis on the run-around heat recovery units that are part of the actual HVAC system in building 99 at the University of Gävle. The HVAC system in building 99 has two separated supply air-handling units in the underground level, each one of them providing a part of the total desired supply air flow. In the same way, there are two exhaust air-handling units in the attic that are interconnected with the supply air-handling units via the pipes of the heat recovery units. As discussed earlier, these units play a very important role in HVAC systems. Therefore, their actual performance is evaluated in order to understand possible improvements to boost their efficiencies.

1.2. Statement of the problem

HVAC devices, not only in Sweden, but also in the rest of the world, today represent the major energy usage service in buildings energy sector. With the current migration of people to cities increasing the need of more and more new big buildings not only residential but especially non-residential (where it is fundamental to control the indoor environment), it urges to find ways to decrease the input energy needs of HVAC systems before achieving the rupture point of the climate.

Heat recovery units that permit the transfer of heat from the extract air to the supply outdoor air, are part of major ventilation systems that is one possible tool to achieve those savings. So, it is extremely important to understand how they are working and how to put them at their maximum point of working efficiency.

1.3. Aim

The main objective of this dissertation is to assess the actual HVAC system in building 99 at University of Gävle, with special focus on heat recovery units in LB20 and LB21, in distinct air-handling units. Based on that analysis, some suggestions to enhance the efficiency of the present heat recovery systems are made taking into account the actual status of the units. Moreover, a superficial economic analysis for the long term, regarding possible savings by improving the efficiency of the systems is carried out.

1.4. Research questions

1. *Why is heat recovery ventilation so important and what are the types of air-to-air heat recovery devices that can be included in HVAC systems?*

2. *How well are the heat recovery units in the ventilation system of building 99, University of Gävle currently working?*

3. *What are the benefits of improving the actual performance of the heat recovery units under analyse and how can that be done?*

1.5. Limitations

This study is limited to two simple data acquisitions, one on a winter day and other on a spring day. Despite this very limited data collection, it is possible to understand how the system works in the most important periods of the year and during most part of it, since the average outdoor temperature in Gävle is of 4ºC and the outdoor air temperatures at the measurement days were about 1,5ºC and 9,5ºC.

The lack of maintenance connected to the bad conditions of both heat recovery units, shows that there is a considerable amount of fouling in the liquid circulating fluid, turning the results in that unit subject of considerable uncertainty.

1.6. Methodology

To carry out the study focused on the efficiency of the heat recovery units in LB20 and LB21, two data acquisitions were done on different days, with a significant outdoor air temperature difference between them. The methodology of this work is essentially based on literature reviews, as well as on the collection of data from both units of ventilation (LB20 and LB21) when a working load close to the maximum, during winter was expected. On a warmer day, some data to access the efficiency was also collected. These records are determined by using a measuring device for liquid medium and also with another measuring device for air medium.

1.7. Chapter overview

This dissertation is divided in seven main chapters, each one of them making reference to a certain part of the study. Chapter 1 describes, among other initial statements, the motivation and objectives of this study. Chapter 2 comprises a general overview of buildings energy, including some justifications for the use of HVAC systems and also a review of its components characteristics. Furthermore, different kind of energy recovery units for HVAC systems are shown. Chapter 3 presents some theoretical basis for the methodology, to understand how to access the efficiency and other quantities as heat transfer for the heat recovery systems under analysis. Moreover, in this chapter, it is also possible to see a complete design description of the HVAC system in building 99. The methodology followed in this research is also displayed. In chapter 4, it is possible to observe the results of the measurements taken in the units that were the focus of this study. Chapter 5 contains the detailed analysis of the data shown in the previous chapter, in order to access the desired parameters. It also contains a superficial analysis about the financial opportunities if the heat recovery units under study are improved. Chapter 6 is the last chapter of this dissertation and includes the main conclusions of the previous study as well as some advices to improve the efficiency of the heat recovery systems and suggestions for further work.

Chapter 2 – Literature review

2.1. Energy in buildings

Inside buildings, energy is used for several different purposes, being the most significant of them heating and cooling, ventilation, lighting and preparation of hot sanitary water. The buildings' sector that includes both residential and commercial, is the largest sector in energy usage among the total share of final energy usage, account with almost 40% of that final amount in the world, for the period 2004-2005 [4]. In Sweden, only the residential and service sectors represents almost 40% of Sweden's total energy use [5]. Heating, Ventilation and Air Conditioning is the principal energy consumer service in buildings. However, the sector of buildings does not comprises only big buildings like public administration and commercial buildings, that require the use of such mechanical systems to suppress the indoor environment needs. Other public services like street lightning, sewage and water treatment plants, power stations and waterworks do not consume so much energy per single unit but they exist in large number leading to a huge total amount of energy use. To understand better how the use of energy was split in Sweden, the *Swedish Energy Agency* analysed and reported in [5] some evidences and conclusions as the fact of more than half of buildings' energy usage is due to heating and provision of hot water, strongly influenced by outdoor temperature. In figure 1, representing the electricity use for residential and service sector in Sweden, is possible to observe an increase in all sectors from 1971 until 1987 followed by stabilization. Regarding to domestic electricity, both the increase in the number of households and appliances on them and the development of energy-efficient appliances, have been working together to maintain the consumption of electricity somehow constant. In the same figure is possible to see that the electric heating suffered an increase until 1988 followed by a decreased in 1997 due to the high electricity prices that leaded to a change to other heat methods as heat pumps, district heating and pellets. The biggest parcel of this figure (owned by business electricity) follows the other two tendencies over the years. This one is a combination of the electricity used by the building (ventilation, lifts, escalators and lighting) and in business activities (computers, appliances and lighting).

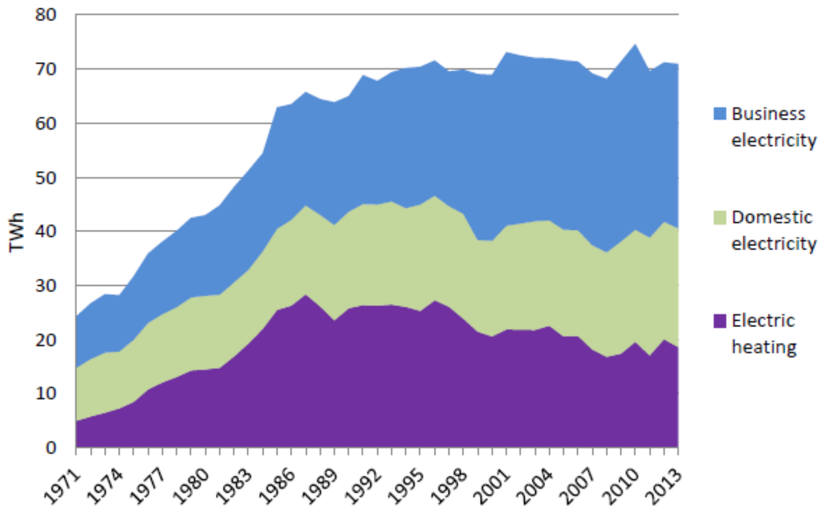

Figure 1: Electricity use in residential and service sectors in Sweden [5]

Most part of non-residential buildings work 365 days per year with a constant need of energy to maintain the desired indoor climate. To understand buildings' energy behaviour during all the year, it is possible to consult diagrams where is shown the number of hours per year of heat deficit and heat surplus. By other words, the number of hours when the building needs an addition or an extraction of heat, using a certain value of indoor temperature as a minimum requirement and taking into account the heat losses (directly related with the level of insulation) and the internally generated heat. In dwellings is not often to find heat surplus enough to require special technical insulation or need of any system to extract heat from inside. In contrast, commercial buildings can have both deficit and surplus of heat that require a heating system and a system to remove the excess of heat. This removal of excess of heat can be an important issue in commercial buildings and is possible to be made by using different techniques like air or cold surfaces. Cooling the space with conditioned air when the incoming air has lower temperature than the air in the room, is regular to see. However, this can increase substantially the air volume, exceeding the normal needed quantity for an acceptable air quality in an office room. Furthermore, in sunny days, big and modern offices can have high amounts of heat surpluses, requiring large airflows to avoid too high room temperatures that will require systems with higher capacity leading to more investments and energy use. When cooling with cool surfaces (e.g. chilled ceiling beams) high temperatures can be avoided independently the current system of air. However, condensation can occur if the cold surface has a temperature below the dew point of the air, which can be prejudicial if the cooling is made by circulation of chilled water in fan-coil units. Despite this problem, cool with cold surfaces usually is a cost efficient solution [6].

Is very important to perform a correct energy balance of a building to understand how the heating, ventilation, and air conditioning system will influence the indoor environment. The heat balance stipulates that the supply heat (by internal generation from people and devices and the heat for space heating) have to be equal to the heat losses (through transmission, infiltration and ventilation).

2.2. Indoor environment

Nowadays is well known that in modern societies, a large part of the population spend more than 90% of their daytime inside buildings, residential or non-residential, where in most part of the cases is possible to find a controlled environment. In indoor environments, people can be affected by different parameters without even realize that. Physical environmental variables as the indoor air quality (IAQ) can be decisive for people' comfort and health, especially due to the resulting consequences like the decrease of diseases and increase of economic and social benefits, immediately or in a long term. Studies have shown that IAQ can affect reasonably the productivity and the level of dissatisfaction on people, in some cases increasing the work performance from 6 to 9% on an office. Those results were obtained when common indoor sources of air pollution (such as floor-coverings), supply air filters and personal computers were removed, or by keeping them in place and increase the rate of supply clean outdoor air from 3 to 10 l per person, instead [7].

Indoor climate is a junction of different variables: physical, environmental, physiological, behavioural and psychological [7]. Thermal variation, air quality, sound, time, stress or light, make part of the space characterisation. Indoor environmental parameters as temperature, humidity and airflow are essential do describe and study an indoor environment. So, to access general indoor environmental quality, is necessary to evaluate the quality of each one of the parameters and see their effect on the physiological sensory system of the human body. Indoor air quality is directly related with the concentration of pollutants in a ventilated space and deserves special attention because it reflects the results of the implication of ventilation systems in buildings. In the next section, the environmental factors of thermal comfort and indoor air quality are briefly discussed.

2.2.1. Indoor air quality

Air quality is strongly dependent not only on the degree to which the air is free from pollutants (what can be annoying or harmful to the occupants), but also on the ventilation rate in the closed space [8]. Indoor air quality refers to the air within the building and around it and its structure, once besides by natural or mechanical ventilation, outdoor air can also penetrate in buildings in "not clear" ways like infiltration (through openings, joints, cracks in walls, floors, ceilings and around windows and doors). Indoor pollution sources are very diverse and have different behaviour in the air. It is important do understand what they are (hazardous or no) and how those pollutant sources act (if they are more or less continuous), in order to estimate their concentrations and the adequate ventilation that is necessary to remove them. Indoor pollutants can be either gaseous or particulates. Among them are dust, fibres, mists, bio aerosols, and gases or vapours. Historically, the main reason for ventilation is associated to the creation of a healthy indoor environment, being the outdoor air supply rate (found by experience) chosen to give reasonable indoor air quality and to prevent the appearance of moisture in building that would degrade it. By coincidence, this supply rate was enough to keep the pollutants concentration below limit levels [7]. Nowadays, is known that the indoor air quality has also influences on performance and productivity of people.

Gaseous pollutants are expressed in mass per volume of air (µg/m^3) or parts by volume, ppm (parts per million) being either organic or inorganic. When considering outdoor air, this kind of pollutants is manly formed due to combustion processes, with natural or anthropogenic causes. However, indoor, gaseous pollutants are created by people, combustion, building materials and furnishings, and products for household cleaning, for example [9]. People are the main source of gaseous pollutants in an indoor space once, when they breathe, they produced large amounts of carbon dioxide (CO_2). Since CO_2 is relatively easy to measure when using the right device, this one is commonly used by many ventilation designers to evaluate if the ventilation is enough to maintain the CO_2 concentration lower than a minimum level of 1000 ppm [10]. Water vapour is not harmful for humans, however, when its concentration is very high, can lead indirectly to health problems, turning the indoor air quality more poor.

Particulate pollutants in air are also worthy of attention. In outdoor air, they result from natural processes (mechanical and chemical processes as erosion and fires, pollens) or human activities (road traffic, industrial emissions, combustion). When analysing indoor air, the main source of this kind of pollutants are clothing and skin. Despite it is possible to remove them almost entirely with resource of filters, it is necessary to take into account their wide range of sizes. Their concentration is expressed in number of particles per volume of air (particles/m^3) or mass per volume of air (mg/m^3 or µg/m^3).

Besides the concentration of CO_2 in a room, there are other indicators of the indoor air quality. The *nominal time constant,* τ_n, can be used as a time constant in a ventilated space, indicating the response time for changes in the air quality. The *local mean age of air,* $\bar{\tau}_p$, is the time required for the fresh air that enters in a room to reach a particular point in it, *p*, and is often considered as a measure of the local air quality, since the longer time the air has been in a ventilated space, more contaminants generated indoor can be assumed be accumulated. It can be calculated by the following formula:

$$\bar{\tau}_p = \frac{1}{C(0)}\int_0^{\infty} C_p(t)\, dt \qquad (1)$$

where *C* is the local tracer concentration and *t* the time.

If the air in the room is perfectly mixed, the average age of it will be equal to the time constant [10]:

$$\bar{\tau}_p = \tau_n = \frac{1}{N} = \frac{V}{\dot{V}} \qquad (2)$$

where *N* is the specific air flow (h^{-1}), *V* the volume of the room (m^3) and $\dot{V}$ the volumetric flow rate (m^3/s).

When considering an entire room and not only specific points, the average age of air, $< \bar{\tau} >$,is the average of the local average ages of the room air and can be quantified by measuring the tracer concentration at the exhaust air point and integrating this in time.

2.2.2. Overall thermal comfort

By definition of ASHRAE [11], thermal comfort is assessed by subjective evaluation and is essentially a *"condition of mind that express satisfaction with the thermal environment"*. According to [6], to the surrounded thermal environment be comfortable at the human perception, it has to follow some criteria:

- The heat loss of the body should be balanced by the heat generation
- The skin temperatures and sweat secretion should be within the narrow limits promoting thermal neutrality, i.e. the person should not want the temperature to be either higher or lower and
- The person should not experience unwanted heating or cooling of a particular part of the body.

The heat balance of a body is strongly influenced by the environmental parameters where the person is, in particular air temperature, relative air velocity, mean radiant temperature of surrounding surfaces and the water vapour pressure in ambient air [6]. The heat losses made through the skin, can exist via different mechanisms such as radiation, evaporation, convection or conduction. Moreover, other characteristics as the metabolic rate of the human body corresponding to the type of activity going on as also the thermal resistance associated to the clothing are important to speed up or retard heat losses, since the clothing acts like a resistance to heat transfer. There are standard values to use to estimate those two parameters that are expressed in met (metabolism rate) and clo (clothing level).

After the estimation and/or measurement of those six parameters, it is possible to calculate two different indices, the predicted mean vote (PMV) and predicted percentage dissatisfied (PPD), that are a good way to express the level of satisfaction about the thermal comfort when the temperature is deviated of its optimal value. PMV index is based in self-reported perceptions by people included in large groups on a sensation scale expressed from -3 to +3 corresponding to different categories, "cold", "cool", "slightly cool", "neutral", "slightly warm", "warm" and "hot" (See Figure 2). By other side, PPD establishes a quantitative prediction of the percentage of thermally dissatisfied people calculated from PMV [11]. An example of those indexes is shown in the figure 2. In the graph, is possible to observe that when values of thermal sensation in the comfort scale are deviated from the neutral point (0), the predicted percentage of dissatisfaction increases.

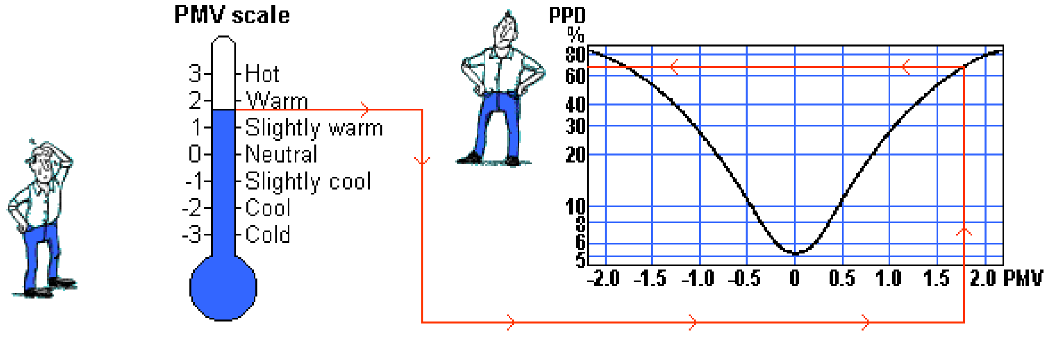

Figure 2: PMV scale in the left and graph with relation between PMV and PPD [12].

The international standard ISO 7730 [13] for moderate thermal environments describes a methodology to analytically determinate and to interpret thermal comfort over the calculation of the PMV and PPD indices and local thermal comfort criteria. Moreover, the standard 55 from ASHRAE also specifies analytic methods to calculate the thermal environmental conditions in occupied places, taking into account the metabolic rate and clothing level of the occupants.

As referred in ISO 7730, it is possible to determine the PMV by different ways as:

a) From an equation developed by Fanger and co-workers (possible to use also in digital computers)
b) From PMV values tables given for a certain activity level and with different combinations of clothing, operative temperature and relative air velocity values
c) By direct measurement, using an integrating sensor (equivalent and operative temperatures)[a].

The PPD can be also calculated by mean of an equation englobing the value of PMV. These two indices (PMV and PPD) are very important to help to evaluate a complex concept as thermal comfort, turning easier and more appropriated the design of HVAC systems. These systems whose main objective is to provide a comfort indoor environment, need to take into account the personal opinion of people because there is not only one single requirement to project a system that satisfy the for a desired indoor environment. Each building is particular and has, for example, its own type of activities going on, leading to different needs.

2.2.3. Requirements in building energy regulations

After the oil crisis of 1970, the idea about the creation of building energy codes turned real in order to aware governments about the need to save energy and create entities responsible to manage energy issues. With the growing concern about environment protection and considering the fact that fossil fuels are "finishing", energy policies are becoming more and more strict. This need of energy rationalization and money savings not only in buildings' area but also in all the other sectors that involve use of large amounts of energy, leaded to new energy regulations and energy certification programmes insert in energy planning policies of each country. Energy regulations exists to establish minimum energy efficiency requirements for the design, construction and retrofitting of new buildings while certification schemes encompass any procedure allowing the comparative determination of the quality of new or existing buildings in terms of their energy use [2].

There are numerous codes or energy standards for buildings. In order to analyse them, is common to find them separated according to the type of building (commercial or residential), size and either if they are service buildings or no. In some countries, despite there is a difference between commercial and residential buildings, there are also specific requirements for buildings like hospitals.

As previously referred, energy requirements are spread over all the areas associated directly or indirectly with buildings that use large amounts of energy. These requirements can be either integrated in general building codes or standards for buildings, or they can be set in separate standards more specialized in other areas like energy efficiency. These requirements are instructions to pursue the reduction of energy use by a process, system or device. In order to achieve success in energy goals for buildings and systems, the fulfilment of the requirements (compulsory by nature) can be proven if the prescriptive is done, by a posterior performance evaluation or by certification of some energy standards (e.g. ASHRAE 90,1 and IECC). When analysing the complete energy system of a building, the different requirements are imposed by the authorities, in some cases local, state, national or supranational as the European Energy Performance in Buildings Directive for Europe and the US based standards (IECC and ASHRAE) used in US and Canada [2].

Due to the complexity of the building as a whole energy system, these requirements can be global (when the analysis of the building is done as one single piece with a limit value for global energy efficiency indicator), of service (when energy intensities of a main building energy services are

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[a]Operative temperature: The uniform temperature of an imaginary black enclosure and the air within it in which an occupant would exchange some amount of heat by radiation plus convection as in the actual non-uniform environment [11].

Equivalent temperature: Similar concept to operative temperature but in this case the effect of a non-uniform environment, as wind velocity, is considered.

limited independently), of demand efficiency (when the total efficiency of an energy system is the ratio of energy demand handled over the energy usage by the system) or low-level requirements (when the consumption of the building is not limited global) [2].

Building energy regulations set up energy codes according to the main energy usage services in buildings (HVAC systems, lighting, sanitary hot water and equipment) and also take into account the envelope of building since it strongly influences energy exchange between indoor and outdoor [2]. For non-residential buildings, it is extremely important to set minimum energy efficiency requirements in building energy codes when analysing HVAC systems once they are the major energy usage service on them. In the following section, the requirements associated to this area are briefly described.

2.2.3.1. HVAC requirements

Heating, ventilation and air conditioning services are part of a complex system with several components, each one of them with a specific function. As it is later better explained in section 2.4 (HVAC systems), these components have different structures and working principles, having each one of them its own energy usage and consequently different opportunity areas for improvements in energy use and working performance. In ASHRAE 90.1 Standard 2004, the requirements for HVAC systems are set for air-conditioners, condensing units, heat pumps, water chilling package, package terminals, room air conditioners, furnaces, duct furnaces, unit heaters, boilers and heat reinjection equipment in individual tables [4].

There are different types of regulations in which requirements are set. To assure that building energy regulations are fulfilled, it is possible to follow one of the two most used ways: prescriptive or performance path. The first one is more strict where rules are over global goals whereas the second one allows other possible approaches to solve the problem requiring only the definition and quantification of the main objective that is, in this case, the reduction in energy usage. In Europe is very likely to find different countries using different approaches.

As referred, the performance path allows more creative approaches to ensure that the buildings' overall usage of energy is below the threshold of the global energy requirement (set based on the supply of energy or the resulting environmental impact), turning everything easier for designers and for the regulatory agency [2]. In this path, free trade-offs can be made between insulation and installation of efficient equipment, but also based on the selection of fuels, the use of renewable energy, the primary design of the building, use of daylight, and intelligent installations or automatics [4]. This option is very important for designers and constructors leading to multiple possible combinations of systems in order to see which is the most economically attractive. Related to this path, two key issues surge, one concerning the estimation of the energy usage and other about how to decide the global requirement, since buildings are all different and have different energy needs. Besides the climate, some parameters to discriminate buildings and set the global requirement of HVAC systems could be building shape, operating hours, building orientation or ventilation rates [2]. There are different methods to estimate HVAC use and most of them involve an estimation of average efficiencies. However, it is possible to estimate reference consumptions by means of simulation programs. Due to this, constructors are required to simulate the building with advanced computer based models, integrating all the different parts and installations of it [2].

Prescriptive path is very important to achieve good results of for example efficiency levels. A possible classification in categories for the different prescriptive requirements in energy regulations is presented in [2]. Those categories are: equipment minimum efficiencies, fluid distribution systems, HVAC control, ventilation, heat recovery and free-cooling. In the European level, some measures like the system of energy labelling of equipment, were introduced in order to inform the consumers, promoting energy savings and energy efficiency and the voluntary certification of HVAC equipment by the adoption of test standards from Eurovent. Not only equipment's have efficiency with minimum values forced by the standards. All the other components that make part of the system have to follow the requirements directed for them and achieve compliance with their specific targets. For example, refrigerant pipes should be insulated with a minimum insulation thickness in order to have an energy efficient system able to transport fluid with reduced heat losses. The specific consumption of fans and pumps is now also a factor that has to be analysed and fit in the standards. All these categories are extremely important to achieve the desired levels of energy usage and a way to keep inside the limit margins the energy consumption. Heat recovery is one of the principal keys to accomplish good results on energy consumption in systems that require large amounts of heat since the reuse of it to preheat incoming fluids represents a big saving. One of the six larger areas of HVAC prescriptive requirements are briefly discussed next.

Ventilation prescriptive requirements

Ventilation is a mandatory requirement to achieve acceptable IAQ, with minimum values for ventilation rate presented in the energy codes of each country building regulations. In [2] is said that those minimum values in the requirements should be considered as a maximum recommendation, since in HVAC systems, ventilation is associated with an increase in energy usage, electric peak demand and operating costs. There are several requirements according to the ventilation issue in question: type of ventilation, minimum ventilation rates, filtration, recirculation and ventilation controls. All the previous mentioned issues of ventilation have their own requirements where is clearly written the situations in which certain measures have to be considered. An example is what are the necessary conditions to allow only the use of natural ventilation in closed spaces, instead of mechanical ventilation. Other important mentioned issue in which such requirements are vital is in how to determine the minimum ventilation rate by indirect and direct methods, explained later in ventilation sub-section.

2.3. Ventilation

The term ventilation can be defined as “the movement of fresh air around a closed space, or the system that does this” [14]. The main objective of ventilation is the removal of generated pollutants and the supply of clean air inside a room, no matter if is through a mechanical mechanism or simply natural (as opening a window). In addition to those factors, ventilation can also be used to provide the space with a supply of air needed to cover the exhaust of ventilated safety equipment as well as a way to remove or supply heat to a room, when it provokes discomfort in the room users. Despite the fact that it removes pollutants generated in a room, it cannot affect the generation of them.

Since the removal of certain gases and particles in indoor air is one important reason to access ventilation needs, it is important to know the rate of pollutants generation to size the airflow ventilation that is adequate to maintain the concentration of pollutants below an acceptable level. Due to that fact and to the requirements for indoor climate, the supply and exhaustion of air will strongly differ from building to building.

The air flow can be measured both in absolute units (m^3/s or l/s) or in specific units (as flows per m^2 of floor area, l/(s.m^2) or in m^3/h per m^3 of room volume). The specific airflow rate is sometimes also denominated as *air change rate per hour (ACH)*, having units per hour (h^{-1}). It can be calculated with the following equation:

$$Air\ Change\ Rate\ (ACH) = \frac{Air\ flow\ rate\ (m^3/h)}{Room\ volume\ (m^3)}\ \left(\frac{1}{h}\right) \quad (3)$$

Considering the main purpose of the implementation of a system and the type of room under analysis, the values of ACH needs can be different. These ones are summarised in appendix A. There are two different ways to calculate the minimum ventilation rate for a space: using an indirect method where the outdoor air volume per person or per unit are is set depending on the activity or by a direct method focused on limiting the concentration of contaminants under certain levels (dilution method, CO_2 based methods and perceived IAQ method).

The air change rate has also associated to it a certain efficiency, usually named as air exchange efficiency, ε_a, and is a measure of effectiveness of air delivery. It is expressed as the ratio of the lowest possible average age of the room air to the real average age of the room air.

In order to avoid a poor thermal climate and big heat demand in a ventilation solution, it is necessary to consider the consequences of having high air change efficiency [8].

The presence of people in a room, the surface temperature of materials, the location of the supply and extract air terminals and the type of activities that are happening in it are important factors to determine how well the air is mixed in certain zones of the room.

2.3.1. Ventilation effectiveness

Considering that two central reasons to install a ventilation system in a building is to remove the pollutants (increasing indoor air quality) and remove or supply heat to the rooms (to have more thermal comfort), it is possible to describe how efficient a system is by completing those tasks taking into account some values associated to each function.

Once analysing the thermal factor, the *ventilation effectiveness for heat distribution or removal*, ε_t, is similar to a heat exchanger effectiveness, taking into account different temperatures. It is expressed as [10]:

$$\varepsilon_t = \frac{T_e - T_i}{T_m - T_i} \, (\%) \qquad (4)$$

In order to measure how effective the ventilation system is in removing internal produced contaminants, the *ventilation effectiveness for contaminant removal*, ε_c, is given by [10]:

$$\varepsilon_c = \frac{C_e - C_i}{C_m - C_i} \, (\%) \qquad (5)$$

Where the subscripts i, e and m refer to indoor, exterior and mean room values of pollutants' concentration. If the aim is to reduce the contaminants concentration to a certain point, it is also possible to calculate the absolute ventilation efficiency, considering the initial concentration and the concentration at the same point after a certain time interval. These calculations should be made when the room is being used since the conditions are very different when it is empty or occupied in normal activities, besides the fact that when the room is occupied, the air tends to become well mixed.

2.4. HVAC systems

To answer the needs of thermal comfort and indoor air quality in a room with occupancy, is necessary to apply some kind of device with capacity to "prepare" and control the supply air volume until the desired characteristics for that zone. Heating, ventilation and air-conditioning is the tool to answer the desired requisites for a certain indoor environment, once it provides proper airflow, heating and cooling to a room. This need of both heating and cooling (not at the same time) from a building is due to large different demands for different spaces. In those systems, the supply air temperature is always lower than the desired temperature for the room.

HVAC system is part of buildings' services. This service is directly connected with two other big and important areas in the building that are the air-handling system and the hydronic heating and cooling system. In turn, air-handling system comprises the services of air distribution system and air treatment system, one of the most important stages of the whole process through which air passes. The air treatment system is directly or indirectly connected with the water distribution systems that are part of the hydronic system due to the presence of heating coils, cooling coils, humidifiers, dehumidifiers and filters.

Despite all the different design possibilities for cooling power, all the systems need to have a way to regulate the supply airflow rate. There are two main ways to control the airflow rate depending on whether the supply airflow volume remains constant or variable.

As its name refer, when a HVAC system is designed as a constant-volume (CAV) system, the supply airflow volume is constant for the whole system, despite the changes in the supply air temperature from room to room. Throughout the system, should exist balancing dampers to control the air flow volume. This type of design is suitable for single zone applications when the heat load is supposed to be constant across the entire space in order to keep the room temperature at a certain value when the heat gain varies, changing for that the supply air temperature [15]. If the CAV system is supplying air for more than one room, there is a risk that the temperature in some rooms be either above or below a desire value. Therefore, in the case where the temperature of the rooms are different and there is need to cool, the supply air temperature has to be enough cold to cool the warmest room. Installed in the air duct of that room, a temperature sensor sends signals to activate the zone reheat coil. In a room that is too cold and does not need more cold air, the sensor will activate the reheat coil, rising the supply air temperature until the desired. This technique is simple but not very efficient for multi-zone applications.

The working principle of systems with variable-air volume (VAV) is very different from the previous one. Here, the supply air temperature is kept in a constant value and the air flow volume changes according to the temperature needed in each room. When there is an increase in the temperature of a room (due to changes in the number of occupants or applications, for example), the sensor used to regulate the supply air flow sends a signal to the VAV box that will rotate a damper in order to provide, in this case, a larger opening for the supply air flow rate, lowering the temperature inside to room until its set point temperature. In the other hand, if the temperature in the room is too low, the VAV box will close the damper plate in order to supress the supply air to that room. This type of design method is widely used in multi-zone applications with different cooling loads once it is an energy efficient system. In the presence of sensors, the temperature is not the only factor taken into consideration to activate the dampers once the concentration of CO_2 in the room is also a determinant parameter in the air flow rate choice.

Unlike big rooms, in some small rooms (similar between each other) it is not possible to regulate the air flow rate, i.e. to increase or decrease the desired amount of air from a specific projected value. Instead of each room have its own control sensor connected to the VAV box, it is often to find only one with it. That room acts as a reference to the airflow volume for the others, whether they need or no more air volume.

There are several possibilities to classify HVAC systems into categories. In agreement to [6], they can be categorised at least in two ways, i.e. according to:

- The type of systems given the heating and cooling media from the plant rooms to the rooms in the building: all-air systems; air-water systems; all-water systems; water-air systems; and unitary refrigerant-based systems.
- Centralized or decentralized system, by other words, if air conditioning is really needed in place of just ventilation: ventilation-only systems; local air-conditioning systems; partly centralised air-conditioning systems; centralized all-air conditioning systems.

2.4.1. General Components

HVAC systems are a junction of diverse components working together to achieve the desirable indoor conditions. Some of those components are air-handling units, fans, ductwork, diffusers and grilles, heat exchangers, chillers and boilers, pumps, controls, among others. This small section provides a brief description of some of the central components in order to allow the reader to understand better the complete system.

2.4.1.1. Air-handling units

The major processes that air suffers are located in this unit. Air-handling units draw the air from outdoor to inside of the building using large fans. Besides the heating and cooling coils, there are also other important components as filters, humidifiers and controlling dampers inside of it, in order to supply the air at the required values of temperature and relative humidity.

The design of this component changes according to the selected model but has to fulfil the requirements in building energy codes or standards. The most famous models for design of such components are the Northern European and the American. Despite both have in common the basic components displaced differently like heat and cool coils, humidifier and heat recovery unit, there is one big difference when it refers to the heat recovery. In the American model, there is not only recovery of the heat, but also recovery of some extracted air, turning the supply air not 100% of outdoor air. The Northern European model stipulates that this mixture between outdoor and extracted air in the heat recovery system cannot happen, since supply air have to be 100% of outdoor air. In the following figure is possible to observe an example of an air handling unit scheme using 100% outdoor air for supply. The fan is usually located downstream of the coils, so the air just passes over it after filtered and passed through the coils that heat, cool and dehumidify the flow (if it needs to be dehumidified). The dampers perform the function of regulate the incoming airflow from outside, opening or closing until a certain angle.

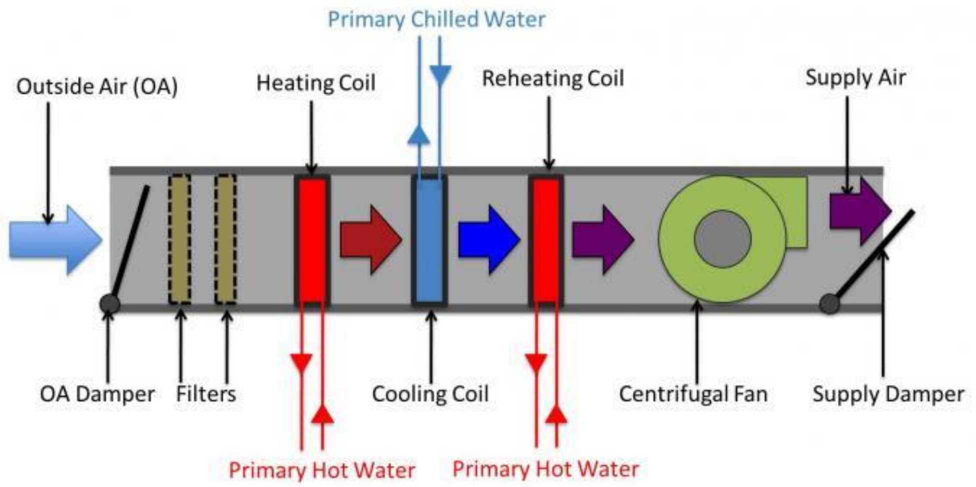


Figure 3: 100% outside air-handling unit [16]

Despite most part of the air handing units are available in sub-assembled sections ready to assemble in the local, there are also air-handling units configured as package units (usually used for small systems with less than 100,000 cfm of capacity). This kind of packages have specified all their components (i.e. fan type, pre and final filter type and size, sound trap type and size and heating and cooling coils size and capacity). These ones can be installed vertically or horizontally, according to the design for that specific place and depending if there is a rooftop application or an application inside mechanical rooms where space is a premium, respectively [17].

2.4.1.2. Fans

Although different, sometimes *Fans* and *Blowers* are used as synonyms of each other since they are similar in terms of circulating and supplying air. However, the big difference between them is the technical part that influences how the air circulates in the system. By definition, a fan is a machine that is used to create flow within a fluid, such as air, whereas a blower is defined as a machine that is used to produce large volumes of gas with a moderate increase in pressure [18].

With object of create a flow of air around and entire room or space, a fan consists in an impeller (containing the blades or vanes and the hub) that rotates in order to direct the air flow, producing air at low pressure, usually by action of a motor which run of electricity.

The fans included in the air-handling units are used to provide force enough to move the air through the system. Since they are essential components of all HVAC, they should be previously selected according to diverse factors, in order to avoid unnecessary inefficiencies, unacceptable indoor air quality and high noise levels.

There are two types of fans: *axial* and *centrifugal*. Each one of them can have its blades moving backward or forward, performing in different ways in each case.

Despite noisier and with low energy efficiency, *axial fans* are very famous in industry's applications for ventilation and exhaust systems, since they can change the airflow velocity, creating static pressure and move large amounts of air at relatively low pressure [19]. This kind of fans can be simple *propellers* or more complex as *tubeaxial* or *vaneaxial.* This difference takes into account some characteristics as the pressure, the number of vanes, the diameter of the hub, the efficiency and whether they have or no guide vanes before and after the wheel to reduce rotation of air stream and recover rotational kinetic energy (in the case of vaneaxial fans). In Figure 4 is possible to see examples of different axial fans.

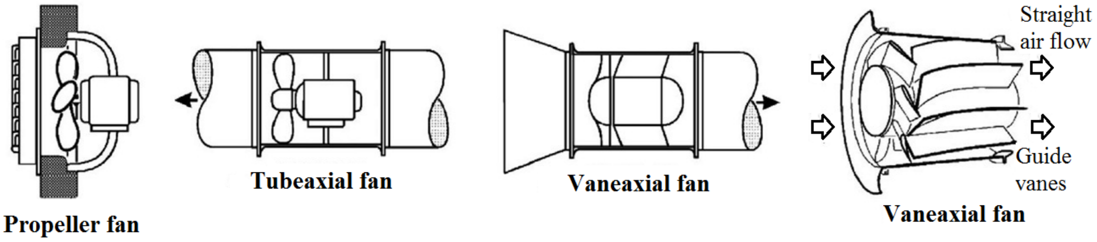


Figure 4: Axial flow fans [20]

With the option to act over a larger range of pressures when comparing to the *axial fans*, the *centrifugal fans* are able to move large or small quantities of air. Figure 5 shows an example of this kind of fans. Due to their capacity to create a high pressure for harsh conditions (as high temperatures or unclean and moist air), they are extensively used in several applications. The principle of work is similar to the centrifugal pump in which a rotating impeller mounted inside a scroll-type housing transmits energy to the air or gas being moved. They can be categorized according to the blade shape that can be radial, forward-curved or backward-curved. Radial-bladed fans are not usually used in HVAC applications, being more common in industrial exhaust applications [19].

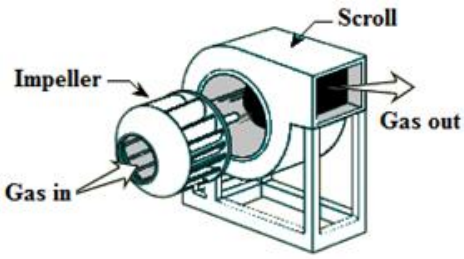


Figure 5: Centrifugal fan

Fan Laws and general performance

Because different characteristics influence the performance of a fan, it is possible to find relations between them. The basic *fan laws* describe the tendency of characteristics as the fan volume flow, the pressure, the speed and power in the system [21]:

1) The volume flow is directly proportional to the fan speed.

$$\frac{\dot{V}_2}{\dot{V}_1} = \left(\frac{n_2}{n_1}\right) \qquad (6)$$

Where $\dot{V}$ is the volumetric flow rate (m^3/s) and *n* the fan operating speed (rpm)

2) The static pressure is proportional to the square of the fan speed.

$$\frac{SP_2}{SP_1} = \left(\frac{n_2}{n_1}\right)^2 \qquad (7)$$

Where SP is the fan static pressure (Pa) and *n* the fan operating speed (rpm)

3) The power required is proportional to the cube of the fan speed.

$$\frac{kW_2}{kW_1} = \left(\frac{n_2}{n_1}\right)^3 \qquad (8)$$

Where kW is the absorbed power (kW) and *n* the fan operating speed (rpm)

As referred before, the performance of the fan is strongly influenced by its characteristics and can be stated in several ways, with the air volume per unit time, total pressure, static pressure, speed and power input. Each fan has a performance curve supplied by manufacturers, in which are represented the required power for the corresponding pressure.

2.4.1.3. Ductwork, Diffusers and Grilles

In order to conduct and deliver the air from the air-handling unit to the conditioned spaces and from these ones to the exhaust air-handling unit, is necessary the existence of a ductwork. The presence of fans in both the inlet and outlet of the system, create a difference of pressure that is responsible for the air movement through the pipes, delivering it in each diffuser at a desired airflow rate (see later VAV and CAV systems). In the total duct structure there are four main sections with different types of air ducts depending on the section: supply air duct, return air duct, outdoor air duct and exhaust air duct. No matter for what kind of system is, the air duct system has many constrains and must consider the following aspects [22]:

i. Space availability
ii. Space air diffusion
iii. Noise levels
iv. Duct leakage
v. Duct heat gains and losses
vi. Balancing
vii. Fire and smoke control
viii. Initial investment
ix. System operating costs.

A poor design of a ductwork lead to systems that run incorrectly or are expensive to own and operate, since there is a big amount of wasted energy and/or ductwork material that increases the life cycle cost of the system. The negative effects are not only economic but is also possible to verify a decrease of comfort and productivity in people.

Since the shape of the air duct influences the pressure losses, it is even more important to do the correct design. Ducts can be rectangular (more easily fabricated on site), round (with less fluid resistance and better rigidity/strength), flat oval and flexible (with multiple layers of polyester film with metal wire or stripes).

Depending on the category of the building, there are certain ranges of pressure differences, between air inside the duct and the ambient air, to respect. Industrial systems can have a pressure different higher than commercial and residential buildings [23].

Duct design

Is very important to use adequate methods to design and size the duct system and avoid unnecessary adjustments to achieve the desired distribution in each room. Proper duct design requires a knowledge of factors that affect the pressure drop and velocity in the duct. There are different duct sizing procedures. According to the SMACNA (Sheet Metal and Air Conditioning Contractors' National Association) and other associations as ASHRAE [22], there are three typical methods used for "non-residential" duct systems that should be considered:

a) *Equal Friction Method*: the main objective of this very versatile method (since it can be used in different systems) is to achieve the same resistance in each path, in order to have a constant pressure loss for the entire system. The proper speed is selected in the main duct closest to the fan, giving a known flow rate that establishes the duct size and the pressure losses per unit length in the main duct. That pressure loss is used as a template for the rest of the system. Since not all ducts have the same length, the shorts will have to be dampered in order to balance the flow rate to each space, which can cause more noise. This method gives an automatic velocity reduction throughout the system and consequently the noise problems. Works both for supply and return. This method is good for VAV systems where balance is maintained by the thermal units [19].

b) *Static Regain Method*: somehow complex and only suitable for supply systems, this method establishes that the recovery in static pressure should be approximately equal to the total pressure loss along the ducts, so at each branch the available static pressure is used to offset the friction loss on the subsequent section of the duct. This is possible due to the decrease of air velocity in the direction of the flow. This method is better to apply in systems with medium to high pressure and for CAV systems with long ducts with many take-offs. It doesn't produce always a completely balanced system since the total pressure is not the same at all branch take-offs. The general procedure is to first select a velocity for the duct attached to the supply, which will establishes the size of the main duct. The first duct to be designed is the one that appears to have the largest flow resistance [19].

c) *Constant velocity:* It is designed to maintain a minimum velocity, in order to carry less dust. The fact that the air velocity is the controlling factor turns possible the prevention of noise due to high air velocity. The duct velocity is determined before the duct size is selected. With the air velocity, the duct diameter size is computed (using the equation of continuity) and the relevant static losses are calculated. Depending on the type of duct (main duct, main branch duct or branch duct) and the type of system (comfort system, industrial system or high speed systems), there are proper values of velocity to use. This method is often used for exhaust systems [24].

Round ducts should be the basis of the design and then, if necessary, converted where is needed. The main reasons of why the duct shape has to be converted are the coordination, cost factors, availability and the initial design in round ducts.

Duct Insulation

The insulation of ducts is fundamental to reduce the effect of heat gains or losses as the air flows through them. Directly related to this heat gains or losses are economic aspects. Is necessary to do the correct planning in order to find the necessary thickness for ducts in that specific system, otherwise, the losses of heat would be high due the poor insulation or the cost would be too high due to the excess of material. It is also possible to have a system not working correctly and delivering the desired heat, since part of it is lost during the way to the diffusers. Depending on the country, different standards state the duct insulation according to the type of building, specifying the minimum value of insulation thickness. The insulation thickness strongly depends of several aspects like the fluid temperature, thermal conductivity of the type of insulation material, size and location of the ducts and pipes [2]. To avoid unnecessary situations as vapour transmission and condensation, it is also often to consider additional insulation and vapour retarders, working or no both in the same system. Duct liners also provide thermal resistance, despite the main purpose of it be the sound attenuation.

To have an accurate design is important to know the duct gains or losses so is possible to calculate the supply air quantity, the supply air temperatures and coil loads [22].

Acoustics

The basis of sound wave propagation requires the existence of three components: source, path and receiver. Sound waves comprise successively compression, decompression and refraction of a medium that can be solid, gaseous or liquid [25]. The propagation of the sound is followed by an energy flux known as the sound intensity. Depending on the sound level, the human ear can receive with more or less happiness the sound, being less sensitive to low and high frequencies and more sensitive do mid-frequencies. The acoustic character of an indoor environment can be defined by the sound level, the frequency distribution of the sound and the reverberation time (as an indirect measure of the attenuation in the space, since it determines the time required for the sound pressure to fall to 1/1000 of the initial value of sound pressure) [6]. Humans are sensitive enough to hear sounds within a frequency range from below 20 Hz up to 20 000 Hz [6]. It is important to have silence environment in spaces as offices, hospitals, hotels or schools and for that is essential to control the noise sources. In HVAC systems, the major sources of noise are diffusers, grilles, fans, ducts, fittings, and vibrations that influences the air velocity in the ducts and the system pressure. The increase of both quantities leads to an increase of unwanted generated noise.

There are some design strategies that can be considered in order to reduce the level of noise like the use of radiused elbows, the change of the position and location of the dampers (to far from outlets), the avoidance of abrupt changes in the layout and the use of large ductwork (result of reduction on velocity) [26]. Without causing significant changes in the performance of the flow and if the changes in the design are not enough to lower the noise levels and achieve the noise criteria in the occupied space, is possible to apply sub-components named silencers in air-handling units or other components. The types of silencers in HVAC are usually also called as sound trappers, chambers or baffles, for example. These units are classified as *dissipative* or *reactive* silencers. *Dissipative silencers* can have different formats and use porous sound absorbing materials to attenuate sound waves, transforming partially the sound energy to small amounts of heat (due to the motion in the fibres during the passage through the material) taking into account the predominant frequency of the sound to choose the thickness of the acoustical linings, as is shown in the next figure. They are usually used in systems with mid to high noise frequencies. The air doesn't suffer significant changes in the direction which leads to an insignificant pressure drop [27][28].

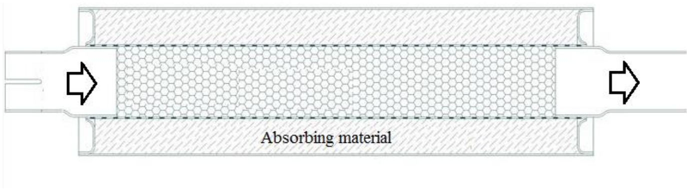


Figure 6: Dissipative silencer [29]

Reactive silencers have a different way of work when comparing to *dissipative silencers,* since they use sound reflections and large impedance changes (by changing the area) instead of absorptive materials to reduce the noise in a pipe or duct [30]. They achieve the sound attenuation with chambers and baffles formed with micro-perforated metal and internal septum's, since the sound waves expand and reflect off the sides of the silencer shell, interfering with the incoming sound waves, reducing the frequency range. They are specially designed to attenuate low frequency noise from machines and can be used in high-temperature and/or corrosive environments. An example of the core of this type of silencer is presented in the following figure.

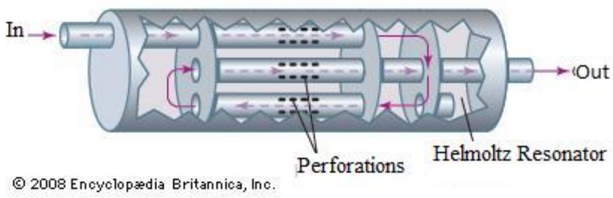


Figure 7: Reactive silencer [31]

2.4.1.4. Heat exchangers

Heat exchangers are widely used and are essential in HVAC systems. These devices are used to transfer heat between two fluids at different temperatures, separated by a solid wall to prevent mixtures between them or direct contact. The heat transfer usually involves convection in each fluid and conduction through the wall that separates the fluids. There are several possible configurations depending on the final application. One design criteria is for example the direction of the flows (parallel to each other or in counter flow).

In HVAC systems, the heat exchangers are embedded in the heating and cooling coils. In the presence of a heat recovery unit, the first heating coil is nominated as preheating coil and the second as reheating coil.

Heating coils are used in the AHU to heat the outdoor air that passes through them during outside cold conditions. There are different types of heating coils used in HVAC applications: electric, hot water or steam.

As the name refers, cooling coils are used to cool the outdoor air that passes through them during warm outside conditions. Besides the cooling capacity, it is also possible to dehumidify the air. In this type of coils, the working fluid is usually chilled water provided by a chiller. It is possible to cool with condensation if the surface temperature of the cooling coil is under the dew point or air (reducing the relative humidity of it) or without condensation if the surface temperature of the cooling coil is above the dew point of air [32].

No matter what is the way to warm up the outdoor air, it is important that the heating or cooling coils have several tubes or resistances made of material with good heat transfer properties, to ensure the maximum possible heat transfer.

2.4.2. Other components in the complete system

The previous mentioned components are probably the most important components in HVAC systems. However, other additional elements complete and help to optimize the system. Some of these complimentary elements are as air separators, expansion tanks, cooling towers, pumps and controls. They can be part of major components or work by themselves, but usually is possible to find them integrated in a major component like for example the existence of a pump, controls and even an expansion tank in the run-around heat exchanger unit.

Air filters

To achieve the desired levels of IAQ, the cleanliness of the supply air is crucial. All HVAC systems should be provided with filters placed in both the supply and exhaust airstreams. Filters reduce the frequency and the level of cleaning, since only they have to be replaced. While the existence of filters in the supply air-handling unit ensures that no insects, dust or leaves enter in the equipment, causing problems and plugging the airflow passage through the coils, the filters in the exhaust air-handling unit are related with the need to ensure that the airflow passage in the heat exchanger is not obstructed by sticky or greasy contaminants or particulates [33].

There are different types of filters, depending on the specific cleanness and efficiency requirements of the ventilated air, being even possible to install basic filters as pre-filters. Some types of filters are: panel type, pocket type, bag type and compact filter. It is also common to find several stages of filtration with different filters, removing in stages the particles with different sizes, from the largest to the smallest. Associated with each filter, there is also a small pressure loss with a certain value given by the manufacturer.

Humidifiers

In cold climates, especially during the winter period, the relative humidity of the air can be relatively low, when compared with its the optimum value for indoor climates. The humidification of the air is a process simple. The addition of water vapour to the air in HVAC can be done by spraying cold water in the AHU or using steam humidifiers. Steam humidifiers are more used due to some characteristics as low hygienic risks, easy mix between dry steam and air and low costs of maintenance [32].

Dehumidifiers

In some hot and humid climates, the relative humidity of the outside air can be so high that is necessary to dehumidify this one until the desired level, before it is supplied to the rooms. To remove the water content from the air in HVAC systems there are some options to consider, being very common to use cooling-dehumidifying coils to condensate the vapour. Other approaches as the use of chemical compounds to absorb moisture from the air are also possible to use [32].

Over the last years, several new experiments have been done trying to develop new dehumidification systems combined with heat recovery, decreasing the use of primary energy in dehumidifier systems [34].

Controls

Controls are one of the most important elements in the HVAC system, being vital to the performance and basic operation of it. They are used in the supply/exhaust air-handling units and in rooms where is possible to track indoor air quality parameters and temperature. By receiving through the local sensors the actual values of the different parameters (as temperature, humidity and pressure), the control system draws conclusions on possible changes that have to be done with base in desired values. Some of those changes can be merely the position of the dampers and valves or for example the speed of the fans. Heat exchanger controls have as main functions the prevention of frost formation and the regulation of energy transferred in the heat exchanger unit under certain operating conditions, according to the type of heat recovery system. Some changes to make in the heat recovery unit to control the amount of energy exchanged are enounced in the next section for the different type of units.

It is also possible to have more control in the function of the HVAC system, using building automation instead of simple direct digital control. Despite involve a HVAC system more complex and expensive, this will allow the owners to control the heating or cooling units [35].

2.5. Heat recovery ventilation

With aim to reduce energy consumption in buildings, more and more heat recovery units have been installed in HVAC systems, more specifically inside the air-handling units. As a rule, heat recovery term is defined as a process of recovering heat (sensible and/or latent) from a flow at high temperature to transfer it to a flow at lower temperature, efficiently and economically viable during the operation.

There are different types of energy recovery systems, like air-to-air or air-to-liquid, but when it concerns to central HVAC systems, air-to-air is the type of energy recovery that is used. Air-to-air energy recovery systems can be categorised according to their method in process-to-process, process-to-comfort and comfort-to-comfort [34]. In process-to-process applications like dryers and ovens, the sensible heat is captured from the process exhaust stream and transferred to the process supply airstream, usually at high temperatures. In process-to-comfort applications as furnaces and paint exhaust, the system heat is captured from the process exhaust and is used to heat building makeup air during winter. By other side, comfort-to-comfort method used in applications like residential, animal ventilation and general exhaust, where the heat recovery device lowers the enthalpy of the building supply air during warm weather and raises it during cold weather, being possible to transfer both sensible and latent heat [33]. Besides the classification by methods, there are several other possible classifications, being the most common according to flow arrangement and type of construction. They are made in several types, sizes, configurations and flow dispositions.

When considering air-to-air heat recovery from the extract air to the outdoor air, it is possible to choose between energy recovery ventilation (ERV), also named as enthalpy recovery ventilation, where there is recovery of both sensible and latent heat, or heat recovery ventilation (HRV), also called sensible hear recovery, where there is only recovery of sensible heat, excluding moisture transfer. Depending on the season, the recovered heat is used to warm or cool the incoming air, being dependent on the climate conditions and the operating period [36]. It is equally important to implement this kind of units both in residential buildings (where the heat recovery can be done using a heat exchanger or a heat pump) and in non-residential buildings. However, due to the fact that non-residential buildings are big consumers of energy, it is much more important to find a suitable heat recovery system to make significant savings. The following presented types of heat recovery devices are mainly used in non-residential buildings.

Despite the fact that heat recovery systems always represent a greater use of electrical energy, the use of such systems will reduce the heat demand for the system, once heat it will be recovered from the extract air that in other way would be wasted to the atmospheric air.

There are two main distinctions about the heat recovery principle when using a heat exchanger: *regenerative* and *recuperative*. *Recuperative* heat recovery means that the heat flows over a heat exchanging surface while *regenerative* heat recovery means that there is a heat transferring mass that is alternatively heat and cooled [8].

Recuperative heat recovery systems can be either indirect or direct. In indirect recuperative systems, shown in the right scheme of figure 8, there are heating coils in the supply and extract airflows relatively far from each other, with an intermediate liquid circulating between the heating coils, preventing any direct contact between air streams. Direct recuperative systems have the heat exchanger positioned between the extract and the supply air streams, with the air flowing through a common piece of equipment as is possible to see in the left scheme of figure 8.

Each heat recovery system that either transfers heat recuperatively or regeneratively, has different possible configurations, characteristics and performance associated to it. In the next sections 2.5.1. and 2.5.2., some examples of the most usual air-to-air heat recovery equipment found in non-residential buildings are shown in the next sections.

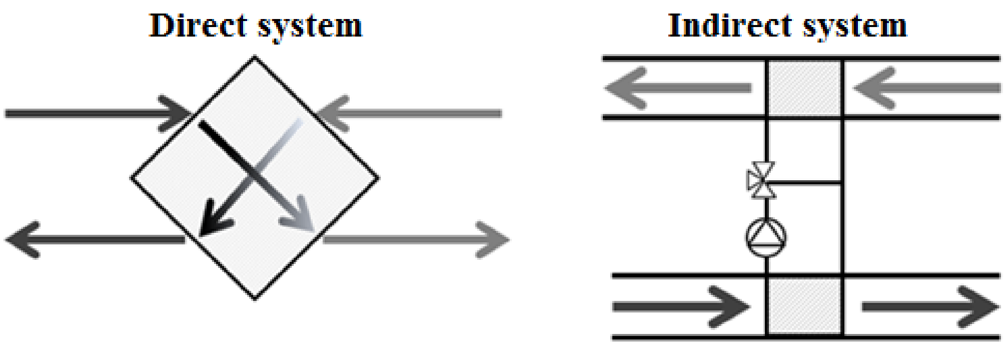


Figure 8: Direct and indirect recuperative heat recovery via heat exchangers

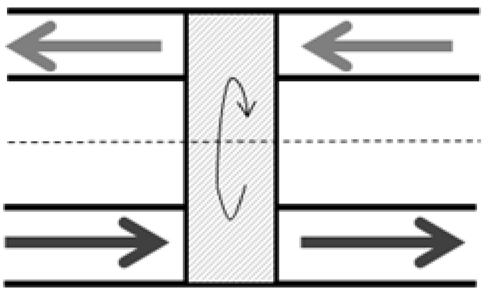


Figure 9: Regenerative heat recovery via heat exchanger

In the last decades, with the emergence of new studies related with the benefits of such units, new regulations, policies, standards and the development of energy-efficiency technologies in this field, made heat recovery systems to become a requirement in building designs mostly in non-residential buildings [34]. There are different requirements establishing the base values from which is mandatory to use heat recovery devices in HVAC systems. According to ASHRAE standard 90.1-2007, section 6.5.6.1., exhaust-air energy recovery is mandatory in cases where the supply airflow is 5.000 cfm (2360 l/s) or more and when the minimum outdoor airflow is 70% or more of design supply airflow [37].

Heat recovery prescriptive requirements

Energy recovery systems are used in a wide range of applications, being more usual to find them in industrial and buildings' sector. This kind of systems is directly connected with energy-efficient measures. The requirements for heat recovery in HVAC systems can be directed to different types of recovery, being some of the most important the flue losses recovery, the exhaust air heat recovery and the condenser heat recovery [2]. It is also possible to find requirements regarding to air and water economizers in HVAC to reduce cooling coil loads, proposing for example the increase of outdoor air volume.

In industries, the recovery of energy from flue gas losses is very usual to see due to significant reductions of energy waste using this one to preheat inlet fluid for boilers. The practise of this technique to preheat an incoming fluid to a boiler or the use of condensing boilers to recover latent heat is mandatory when it refers to flue losses recovery. The use of condensing boilers (inside of heat generators) in HVAC system is also an option when is desired to recover both sensible and latent heat.

In HVAC applications, is very common to see heat recovery from exhaust airflow. Similarly to the previous case, here the extracted heat is used to preheat/precool the outdoor incoming airflow to the system. Since the energy recovery depends on several aspects of the installed system, energy regulations frequently specify under which conditions this type of technology is mandatory, referring the supply air flow volume, outdoor air ratio, operating hours or exhaust air volume, and also the minimum efficiencies for such devices. Other codes only take into account the supply or the exhaust air volume to require the use of a heat recovery system [2]. Due to changes in the system when installing heat recovery devices, there is the need to create a bypass circuit to allow an effective operation of the economizer since this one will keep the energy recovery system from preheating the outdoor air when the air economizer is operating [2].

In zones where there is excess of heat and there is need of cooling, it is possible to condense heat using cool generators. Condenser heat recovery is desired to meet sanitary water heating demand or simultaneous heating and cooling loads in HVAC. ASHRAE 90.1 and IECC require condenser heat recovery for preheating service hot water in large 24h facilities [2].

2.5.1. Enthalpy recovery ventilation

As mentioned before, enthalpy heat recovery ventilation is a type of heat recovery ventilation with transfer of both latent heat (i.e. moisture) and sensible heat (i.e. temperature) from one air stream to the other.

This type of devices is recommended to use in areas where is possible to find one of these two different climates: hot, humid climates (where moister transfer from supply air to the exhaust airstream reduces air-conditioning loads) or in cold, dry climates (where moisture transfer from exhaust air to the supply airstream reduces humidification needs) [33].

2.5.1.1. Rotary wheel

With capacity to recover both sensible and latent heat, this heat recovery device has been largely used in HVAC applications especially in places where the relative humidity of the air is not inside the ideal range of values. When the focus is to transfer not only sensible heat but also moisture, it is first choice in humid climates, since it can improve humidity control by removing the water vapour from the incoming outdoor air and shifting it over the exhaust air in summer as well as remove it from the exhaust air and transfer it over to the outdoor air (eliminating the need of an additional humidifier component) in winter. It is also used for humidity treatment in buildings that easily get damaged with excess of humidity. This is possible using a medium that is covered with desiccant (or absorbent) material, like molecular sieve, silica gels, activated alumina, etc., that concedes moisture recovery characteristics [38].

The operation principle is based on energy transfer between two counter flow air streams in adjacent ducts, one carrying extract air and other outdoor air in counter-flow. The heat is transferred by mean of a rotatory wheel filled with an air-permeable medium that provide either random or oriented flow through the structure, hold by frame construction. Sensible heat is transferred by the medium filled in the rotatory wheel that stores heat from the heat airstream and discharges it in the cold airstream. The latent heat transfer can be either due to condensation (the medium condenses moisture from the airstream with higher humidity ratio) or evaporation (is the medium releases the moisture through evaporation and heat pick up into the airstream with the lower humidity ratio) inside the wheel. Thus the moist air is dried while the drier air is humidified [33]. The passage of heat from the hot upper part of the wheel to the lower part is presented in the next figure.

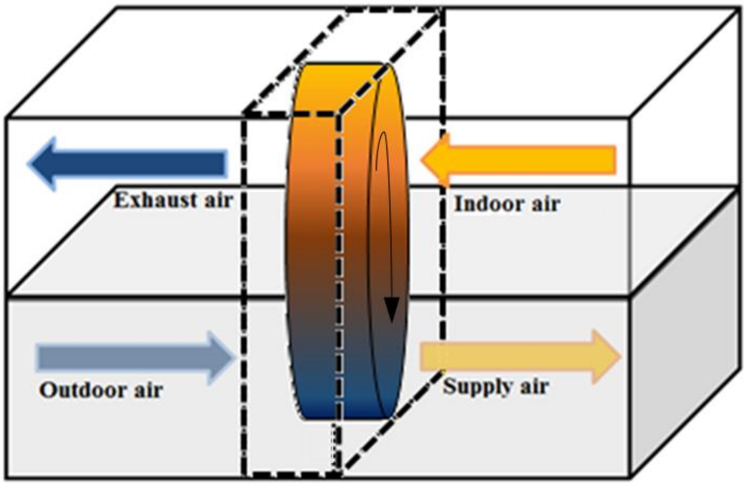


Figure 10: Rotatory wheel scheme for heat recovery system

Different aspects like the dew point and air properties of the extract and outdoor streams, have to be considered when choosing the materials for these devices. The media surface have to be chosen also considering if the final heat transfer is only sensible heat or also latent heat.

The speed of rotation is relatively low, between 3 and 15 rpm and controllable [34]. Due to this possible change in the velocity of the rotor, the efficiency of such device can also be different in time. By doing some investigations in this type of energy recovery unit, it was shown that the energy recovered by the rotatory wheel was around 2.5 times bigger than the heat recovered by a sensible heat exchanger, being this one also the most efficient solution when it turns about recover water vapour energy carried from ventilation and remove the excess of it within the fresh air supply, among all the available options [34].

There are two types of rotors regarding to the type of energy transfer in this regenerative heat recovery device: *Non-hygroscopic* when there is only transfer of sensible heat and *hygroscopic* when there is both transfer of sensible and latent heat. This device can be also named as enthalpy exchanger and nowadays the heat wheels with this transfer property have been used not only as enthalpy recover system but also as desiccant wheels for humidity treatment for dehumidification [38].

The main disadvantage of this type of heat exchanger is the possibility of cross-contamination between exhaust and outdoor air streams due to the rotation of the wheel. Cross-contamination can come from leakages and/or carryover. Leakage contamination happens because of the air passage through seals or gaps due to lack of tightness between adjacent surfaces or because of a differential of static pressure between airstreams, whereas carryover leakages are produced when a small fraction of exhaust/inlet air is trapped in a passage and is switched from exhaust to inlet or vice versa [39][33]. This possible contamination impedes the use of this kind of heat recovery units in HVAC systems where there are toxic, hazardous or delicate functions going on. In order to avoid pollution and transfer of odours to the inlet air, it is possible to install a purge section in the wheel. The purge zone utilizes the pressure difference which exists between the outdoor and return air streams (being the air pressure of the returned air lower than the outdoor air) to "purge" the transfer media with clean outdoor air prior to its rotation into the supply, acting like a bypass section where a small amount of supply air is bypassed through the section before the main supply air flow goes through the wheel [40][38]. The fact that both the supply and the extract air hand-handling units have to be together (since they share a common component) and the additional need of electricity supplied to the rotor, can also be considered other disadvantages of these devices.

In cold climate conditions there is possibility of frosting, however not very common since frost process is very slow, taking a few hours do grow. In the case where frost formation happens, it may be necessary to heat the supply air, stop the wheel, or, in the case of small systems, use a defrost cycle for frost control [33]. Like all the heat recovery devices, this one also should be inspected and cleaned regularly since there is always possibility of passage of dust or other particulates.

The typical sensible effectiveness of this device when the supply and the exhaust airflows are balanced is from 50 to 80% for sensible heat transfer and 55 to 85% for total heat recovery [33].

2.5.2. Sensible heat recovery ventilation

In sensible heat recovery devices, the only parameter that is "exchanged" between airstreams is the temperature. Moisture transfer does not occur in such type of devices despite be possible to have latent heat transfer into the same airstream if the warmer airstream is cooled below its dew point and condensation occurs (latent heat of condensation).

Devices with these properties should be used where latent heat transfer without moisture transfer is desired.

2.5.2.1. Fixed plate

Air-to-air heat recovery by mean of plate heat exchangers is classified as direct recuperative system having this designation due to its construction. Since it is a direct system, this commom unit is positioned between the supply and extracted air streams, with both flows passing through it, however, not mixing with each other. This type of heat exchangers is widely used since it presents high possibilities to exchange heat, being the typical sensible effectiveness between 50 and 80% [34]. Since the two airflows are completely separated, no moisture is transferred, despite being possible to transfer latent heat of condensation (from moisture condensed as the temperature of the warm airstream drops below its dew point) [33]. There are two possible arrangements of the unit to allow the passage of the airflow: several thin plates stacked together or an individual solid panel with several internal airstreams, providing a large heat transfer area. The material of such plates may differ from polypropylene to aluminium, changing also the properties of the heat transfer.

The working principle of this air-to-air heat exchanger is based on the sensible heat transfer between heat exchange surfaces, from the outgoing air to the incoming air stream, as shown in Figure 11. The airflow arrangements can also differ according to the option of design for a certain place, being possible to have counter-flow, cross-flow and parallel flow as alternative.

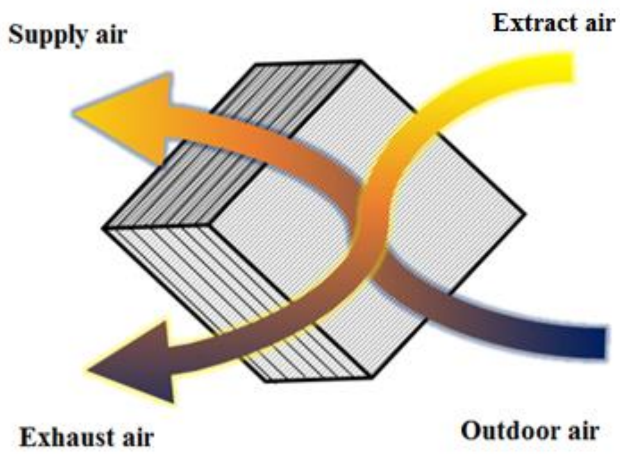


Figure 11: Core of a cross-flow fixed-plat heat recovery system

Due to its high efficiency and low installation, operation and maintenance costs, it is widely used over the world. Since there are no moving parts as there is for example in the rotary wheel, there is no need of regular maintenance of cleaning and calibration.

Sometimes, due to the possibility of frosting and condensation, it is preferable that the returned air passes through the heat exchanger from top to bottom, turning easy the removal off the condensate. At low outdoor temperature and if the exhaust returned air has a temperature lower than the dew point, the water vapour condensates on the surface of the heat exchanger. Then, if the temperature of the plate surface is negative, ice is formed on it, increasing the pressure drop and decreasing the efficiency of the unit [38]. There are some different ways to prevent the condensation and frost in this heat recovery unit such as a bypass channel that conduct the outdoor cold fresh air around the heat exchanger when its temperature is lower than a certain value. These bypass channels, with regulated dampers integrated, placed in the middle of the two blocks of the exchanger, allow a complete or partial bypass of the outdoor air around the heat exchanger not only because of the frost but also to regulate the device.

The risk of cross-contamination by leakages is approximately 0 if the different pressure between air streams is negligible. However, if the air velocity increases, this pressure difference will increase, leading to possible problems of leakages.

New studies have been made in order to improve even more the recovered amount of heat in these devices, like the effects of the membrane spacing and thickness and the creation of new types of porous membranes that allow not only the transfer of heat, but moisture too, providing also enthalpy exchange and increasing the typical effectiveness of total heat transfer to 55-85% [33].

2.5.2.2. Heat pipe

A heat pipe is a passive heat recovery device that is essentially a hollow cylinder filled with some vaporizable liquid acting as heat transfer fluid between the outdoor and the extract airstreams. Usually, heat pipe heat exchangers comprise several very thin copper tubes closed in the tips in which a fluid is continually evaporated and condensate by the extract and the outdoor air, respectively, without need of an external source of heat to allow the phase change. The latent heat of vaporization is used to transfer the heat over the tube. The difference of densities due to the phase change makes the heat transfer fluid to move from one side of the pipe to the other, in order to deliver/receive heat. It is even possible to control the slope or tilt to control the amount of heat transfer and with that reverse the direction of the heat flow, modulate the effectiveness to maintain the desired supply air temperature or to decrease effectiveness to prevent frost formation at low outdoor temperatures [33].

It is important to have a working with high latent heat of vaporization, a high surface tension, and a low viscosity over the temperature range, being thermally stable at operating temperatures [33].

This heat exchanger can be used both in winter or summer. During the winter, the heat provided from the extracted air warms the working fluid, evaporating it inside the closed heat pipe and condensing in the other side of it, when the heat is given to the incoming outdoor air. In summer, the cycle is reversed once the incoming outdoor air is pre-cooled in the entrance, being the sides of evaporator and condenser inverted from the ones in figure 12. In tropical climates, where the outdoor temperature and relative humidity are high, this type of heat recovery system is highly recommended to install. It has taken its place in dehumidification system since it can remove a large part of vapour water from the outdoor fresh air [34]. This type of heat exchanger can be installed admitting one of the following options: Flat, vertical or horizontal.

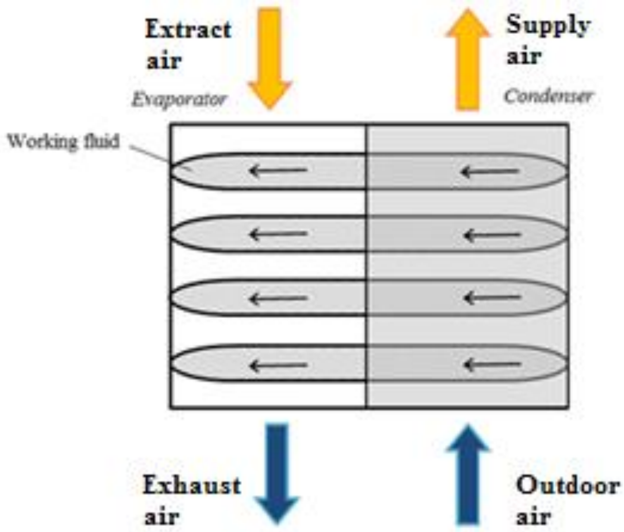


Figure 12: Heat pipe recovery system for winter operation

Since the heat is transferred through a sealed pipe without direct contact within the airstreams, there is no risk of cross-contamination. Generally, HVAC systems with this kind of heat recovery system have the exhaust and fresh air flowing in opposite direction. Despite the fact that sensible effectiveness of heat pipes is not very high (45%-65%), there are other advantages related to its use as the absence of moving parts, the flow resistance, no external power requirement, full reversibility, no cross contamination, compact and suitable for all temperature applications in HVAC and easy to clean [34]. Like in other systems, the efficiency also depends on the heat exchange area that here is defined by the number of rows. Perhaps the main disadvantage of it is that both the air-handling unit and the return air unit have to be placed together since they share this common component. If the velocity of the air flows is very high or if the thermal contact between fins and pipes is poor, the effectiveness of it rapidly decreases.

2.5.2.3. Run-around

This kind of heat recovery device is an indirect recuperative system once is it based on two physically separated heat coils connected via an enclosed solution that transfers heat between them. The energy transfer is reversible for winter and summer, preheating or precooling the outdoor air, respectively. In Figure 13 is possible to observe the operation principle of this type of system.

As referred, the finned tubes that establish connection between the two coils, contain a heat fluid agent properly selected that can be for example water. However, in order to reduce possible problems of freezing, it is common to add a percentage of an anti-freezing substance like *glycol*, creating a new solution with new properties of freezing point, specific heat and viscosity. To host possible expansion or contraction of the heat transfer fluid, it is also required the existence of an expansion tank connected to the circuit. One advantage of systems with circulating fluids, is that there is no humidity transfer between the supply and the exhaust air streams [38]. The pipe system must be insulated based on the actual temperature conditions in and around the system and is fitted with a pump, a control and a monitoring equipment. The control unit receives signals from the duct sensor in the outdoor air and controls the three-way valve (that is usually automatic) in order to ensure that the temperature of supply airflow has a certain value. In certain weather conditions, there is a risk of freeze up the exhaust air cooling coil, so the duct sensor regulation forces the motor of the 3-way valve (with temperature sensors) to the position in which the heating coil is bypassed [41].

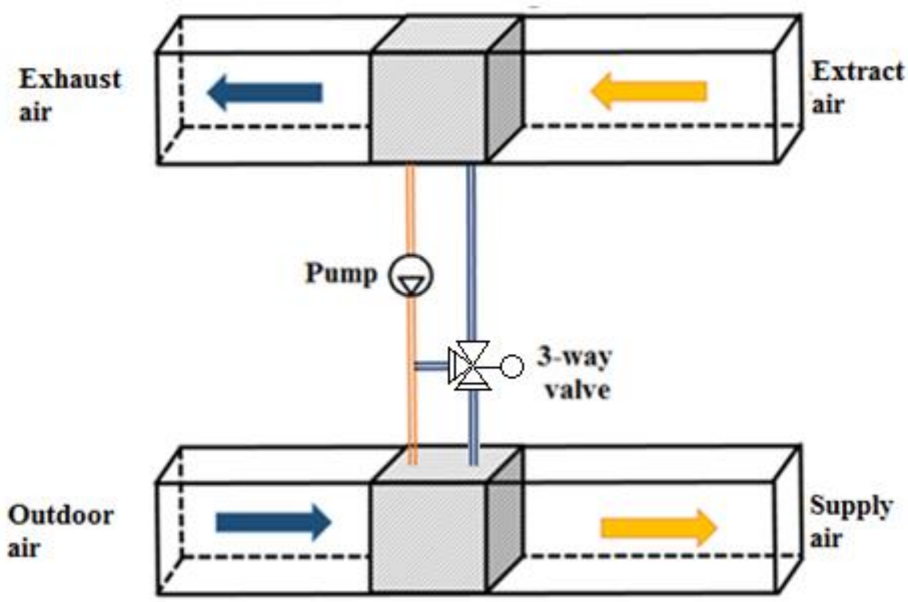


Figure 13: Run-around heat recovery system

Since the coils are physically separated, this kind of system is widely used in situations where it is inappropriate to combine the supply and the exhaust air-handling units in one single section, being even possible to have several meters of height difference between them. Given that the heat transferred between coils is made by a circulating fluid restricted to the pipe system, there is no possibility of contact and cross-contamination between air streams. This induces to no leakages between both fluids.

The main disadvantages of this system are related to the existence of an intermediate circulating fluid. There is a need to have a pump to ensure the loop, increasing the use of electricity and furthermore the efficiency of the system is reduced when using a heat transfer medium like an ethylene glycol solution that has higher viscosity of water and less specific heat. If changes in the flow of this heat-transfer agent are made, like shunt or reductions, the efficiency of the heat recovery unit will change [42]. To ensure optimum operation, the air should be filtered, the coil surface cleaned regularly, the pump and valve maintained, and the transfer fluid refilled or replaced periodically [33].

The efficiency depends on several factors as for example the size of the heat exchanging surfaces. For this type, the sensible effectiveness is between 0.55-0.65 [33], varying among studies. The saving of sensible heat during the winter are much larger than the ones verified during summer, so, the insertion of a run-around cycle is justified based on winter operation.

In the past years, more and more studies were made in order to understand and optimize this type of heat recovery system. According to a review study on heat recovery technologies for building applications [34], at a constant NTU, the system has its optimum performance when the heat capacity rates of the air and coupling liquid were equal. It is also said that the performance of a run-around coil system is sensitive to the rate of circulation of the secondary fluid and that the optimum rate change with changes in the flow rates of the primary fluids or in the fouling resistances.

This variety of heat recovery device is the adopted system in building 99 in the University of Gävle, being discussed further down in next chapters.

Ethylene glycol

This compound with molecular formula $C_2H_6O_2$ and molecular weight of 62.06784 g/mol, is a very important colourless, odourless, viscous dihydroxy[b] alcohol. It is soluble with water and for the most part with alcohol and other organic solvents. Its melting point is at -13ºC and boiling point at 198ºC. Glycols are usually stable if the ambient conditions are between normal values. The freezing point, similarly to other properties as boiling point, specific heat capacity, viscosity and specific weight of ethylene glycol and water solutions, change according to the percentage of it in the solution. In the following tables are presented some of those properties changes of water and ethylene glycol solutions.

The addition of glycol to water yields a solution where is possible to have lower temperatures without instantaneous freezing problems, as would be seen for water. For a solution with a percentage of glycol, the viscosity increases with the decrease of temperature until a critical point where it just stops to flow. It is important to have a low freezing point not only because of the circulation of the fluid but also to avoid rupture or burst due to changes in the volume of the solution, since there is a possibility of fluid's expansion in the confined system if the temperature continues to decrease and ice crystals begin to form.

Table 1: Freezing point of an ethylene glycol water solution [43]

Freezing point

Ethylene Glycol solution (% by volume)		0	10	20	30	40	50	60
Temperature	(ºF)	32	25,9	17,8	7,3	-10,3	-34,2	-63
	(ºC)	0	-3,4	-7,9	-13,7	-23,5	-36,8	-52,8

The amount of heat that is required to raise a unit weight of a substance one degree in temperature is defined as the specific heat. Since the specific heat of water is 1 at standard conditions, it is expected to have in increase of specific heat when more water is added to the glycol solution.

Table 2: Specific heat of an ethylene glycol water solution [43]

Specific Heat - C_p - (Btu/lb.oF)[c]

Temperature		**Ethylene Glycol Solution (% by volume)**						
(ºF)	**(ºC)**	**25**	**30**	**40**	**50**	**60**	**65**	**100**
-40	**-40**	1)	1)	1)	1)	0,68	0,703	1)
0	**-17,8**	1)	1)	0,83	0,78	0,723	0,7	0,54
40	**4,4**	0,913	0,89	0,845	0,795	0,748	0,721	0,562
80	**26,7**	0,921	0,902	0,86	0,815	0,768	0,743	0,59
120	**48,9**	0,933	0,915	0,875	0,832	0,788	0,765	0,612
160	**71,1**	0,94	0,925	0,89	0,85	0,81	0,786	0,64
200	**93,3**	0,953	0,936	0,905	0,865	0,83	0,807	0,66
240	**115,6**	2)	2)	2)	2)	2)	0,828	0,689
280	**137,8**	2)	2)	2)	2)	2)	2)	0,71

1) Below freezing point 2) Above boiling point

[b] Molecule containing two molecules of hydroxyl (OH) radical

[c] 1 Btu/lb.°F =4186,8 kJ/(kg.K)

Due to the fact that solutions of water and ethylene glycol present a lower specific heat value than clean water, when using this kind of solutions for heat transfer, the circulated volume must be bigger compared with a system only with water, in order to circulate more fluid and transfer the same heat as if the circulating fluid were water.

Ethylene glycol (see Figure 14) is largely manufactured due to the high necessity of products containing it. Due to its chemical and physical properties is commonly used as antifreeze in cooling and heating systems, in hydraulic brake fluids, as industrial humectant, as ingredient of electrolytic condensers, as solvent in paint, in plastic industries and synthesis of safety explosives, among others. Regarding to the toxicity associated to it, is labelled as a carcinogen[d] with negative effects and damages to the body when this one is exposed to it, ingested or inhaled. Since it is liquid, it can easily penetrate the soil and contaminate the groundwater. Due to this, it should be avoided if there is a slightest change of leakage to potable water or food processing systems, being preferable to use solutions based on propylene glycol. Polyethylene glycol polymers are formed by the reaction of ethylene oxide and water under pressure in the presence of a catalyst and are chemically stable in air and in solution [44].

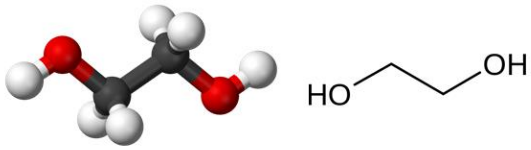


Figure 14: Chemical structure of ethylene glycol molecule [45]

Propylene glycol

Propylene glycol (see Figure 15) is a liquid synthetic compound with molecular formula $C_3H_8O_2$, molecular weight of 78.09442 g/mol and two hydroxyl (OH) groups. Since it has the property of absorbing water, it is commonly used in chemical, food and pharmaceutical industries, absorbing extra water, maintaining moisture levels in certain medicines, cosmetics or food products [46]. It is also used as a solvent for food colour and flavours, and in paint and plastics industries or to create artificial smoke or fog, for example. This clear, odourless, tasteless and colourless compound may also exist in air in vapour form, since it is completely water-soluble. Its boiling point is at 187.6ºC and melting point at -60ºC. This compound is stable at cool temperatures when in a well-closed container but when it is opened to the air, at high temperatures, it tends to oxidize and to transform in other products.

Similarly to Ethylene Glycol, some physical properties of this compound also change with differences in its concentration on water. These differences are shown in Table 3 and Table 4.

Table 3: Freezing point of a propylene glycol water solution [47]

		Freezing point						
Propylene Glycol solution (% by volume)		**0**	**10**	**19**	**29**	**40**	**50**	**60**
Temperature	(ºF)	32	26	18	7	-8	-29	-55
	(ºC)	0	-3	-8	-14	-22	-34	-48

[d] A substance or agent that causes cancer

Table 4: Specific heat of a propylene glycol water solution [47]

Specific heat C_p							
Propylene Glycol solution (% by volume)	0	10	20	30	40	50	60
Specific heat –Cp- (Btu/lb.°F)	1.000	0.980	0.960	0.935	0.895	0.850	0.805

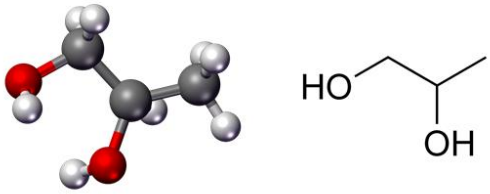


Figure 15: Chemical structure of propylene glycol molecule [48]

Corrosion and degradation

Heat recovery systems or any other system that uses heat transfer fluid based in glycol solutions, requires special attention when it concerns to material selection and maintenance. Under normal storage conditions and absence of air, glycol is quite stable. In the presence of air and even more when heated, it is highly probable to occur oxidation with creation of oxidation products as carbonyl compounds and acids [49]. Oxygenation and heat are not the only means to degrade glycol. Biological activity is also possible to occur when the percentage of glycol is under a certain value. These three means of glycol degradation are connected between them with greater the operating temperature and greater the aeration of the system, greater the degradation of the glycol.

In cases where oxidation is avoided and controlled, it is possible to have systems with heat transfer fluids based in glycol solutions lasting more than 20 years of reliable service [50]. However, due to possible mistakes done primarily in the design or during the operation period (like bad maintenance), some problems begin to emerge and rapidly grow until uncontrollable levels of corrosion, degrading quickly the system and decreasing its performance. In the design stage, the choice of the construction material is extremely important not only to ensure quality of the product and prevent leaks to the environment though valves, seals or defective fittings, but also to secure that both the material of the system and the heat transfer fluid are compatible and not very likely to corrosion.

In order to have further assurance that the glycol system is adequately protected against corrosion and glycol degradation, some inhibitors and buffers must be added to the solution when they do not come already premixed. Generally, inhibited glycol (either ethylene or propylene glycol) contains four to six percent of inhibitors [50]. If no buffer is joined, the degradation of glycol will form organic acids that will reduce the pH and increase even more the corrosion.

Chapter 3 - Methodology

To analyse how an installed HVAC system is working, some data has to be collected and properly analysed. Some sensors of temperature, pressure and fan load are strategically placed both in the supply and exhaust air-handling units to transmit their values to some place where is possible to control and track automatically the system via a software. There, those values are shown to the operator in order to visualize them and identify possible anomalies. However, these sensors are not enough to do a complete performance study of the system, in special of the heat recovery unit in it. It is necessary to prepare a quantitative research method to access the performance of those units. Besides the previous mentioned sensors, there are different types of valves placed along the water pipes in the supply air-handlings unit that are prepared for possible punctual acquisition of values through the insertion of a matching sensor. In those points (in each balancing valve), pressure difference, temperature and flow of water or water glycol solution are collected with the use of TA-SCOPE that is also presented in this chapter. Temperature and velocity of the incoming outdoor air before the heat recovery unit and after it are measured with a hot-wire anemometer, as also happens in the extract air-handling unit for the extract air.

After the manual data acquisition, all the calculations were performed in Microsoft Excel.

3.1. Heat recovery units related calculations

3.1.1. Efficiency

To access the efficiency of both heat recovery units, is important to note that this one depends on several factors like the type of heat exchanger, the size of the heat exchanging surfaces and the heat transfer properties of the material surfaces.

There are significant differences on the principle of heat recovery when the same is done by mean of a heat exchanger or by recycling returned air. If the exhausted air is recycled by using a damper, the supply air will be mixed with a portion of extracted air. The temperature efficiency of this kind of system can be changed by regulating the dampers on the ducts. When the temperature of the mixture of return air and outdoor air rises above the required supply air temperature, the quantity of returned air has to be decreased to a certain value.

The efficiency of a heat exchanger is defined as the "temperature change on the supply side divided by the maximum available temperature difference". In the European standard EN 308:1997 the efficiency is called "Temperature ratio". Admitting heat recovery with heat exchanger without using returned air and considering a balanced system (where the inlet air flow is equal to the outlet air flow), the temperature ratio of the heat recovery system is analysed considering the temperatures in the inlet and outlet of it, being defined by the following equation [51]:

$$\eta_{Tb} = \frac{T_{rec} - T_e}{T_{ext} - T_e} \tag{9}$$

The air temperature after this one be pre-warmed in the pre-heating coil will be:

$$T_{rec} = T_e + \eta_T \cdot (T_{ext} - T_e) \tag{10}$$

If the system is unbalanced, the ratio between the supply and the exhaust airstream should be considered and used as a multiplying factor in equation 9. The temperature ratio will be:

$$\eta_T = Ratio \cdot \eta_{Tb} = \left(\frac{\dot{V}_e}{\dot{V}_{exh}}\right) \cdot \left(\frac{T_{rec} - T_e}{T_{ext} - T_e}\right) \tag{11}$$

ASHRAE 84 standard gives another definition for efficiency of a heat recovery system and instead of define a temperature ratio, it states "effectiveness" as being "the actual transferred quantity of energy divided by the total possible quantity of energy that could be transferred", similarly to the concept of temperature ratio. For that, is also defined a ratio between the supply airflow, $\dot{V}_e$, and the minimum airflow between the supply and the extract streams, $\dot{V}_{min}$:

$$R = \frac{\dot{V}_e}{\dot{V}_{min}} \tag{12}$$

So, the sensible effectiveness is calculated by applying the following equation:

$$\varepsilon = R * \left(\frac{T_{rec} - T_e}{T_{ext} - T_e}\right) \tag{13}$$

These equations are interesting to understand the influence of the airflows on the amount of heat recovered, since most part of the systems are not balanced, i.e. the supply airflow is different of the exhaust airflow.

Similarly to ASHRAE 84 standard, AHRI Standard 1060 concerning to the performance rate of air-to-air heat exchangers for energy recovery in ventilation equipment [52], also defines the calculation of the effectiveness instead of temperature efficiency.

The concepts of efficiency and effectiveness are different from each other. In heat recovery ventilation, the efficiency represents the percentage of heat that the heat recovery unit have recovered while effectiveness represents the percent of heat recovered compared with all the heat which could possibly be recovered from the outgoing airflow.

In the next figure is possible to observe a basic scheme of a ventilation system, with both supply and extract air-handling units and a heat recovery unit making the connection between them, in this case transferring heat from the top to the base.

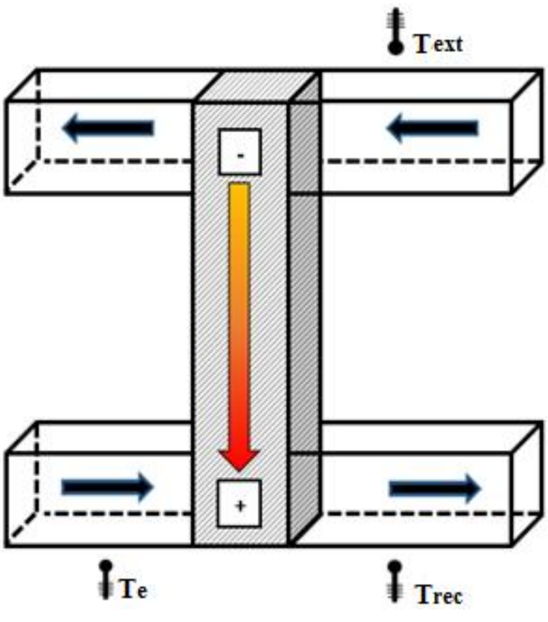


Figure 16: Scheme of heat recovery using heat exchangers

In the case of heat recovery with resource of a heat exchanger, there is always 100% of supply air coming from outside, making the temperature efficiency dependent of the outdoor temperature. If the supply air temperature after the pre-heating coil is higher than the required, the recovery efficiency should be reduced. There are different ways to change the value of efficiency taking into account the type of heat recovery system. In the case of a system with a run-around heat exchanger, the efficiency of the system can be reduced using a three-way valve that creates a bypass circuit for the circulating fluid after the extract air coil.

3.1.2. Heat transfer

The principal purpose to install a heat recovery unit in a HVAC system is to recover heat with as small temperature loss as possible, otherwise, it would be necessary to supply even more power to the system. In order to understand how large is the impact in energy savings, the energy associated to each coil should be find.

The maximum possible rate of sensible energy transfer by a heat recovery unit is defined by [33]:

$$q_{max} = \dot{V}_e \cdot \rho_{air} \cdot C_{p_{air}} \cdot (T_{ext} - T_e) \text{ [kW]} \tag{14}$$

Without any type of heat recovery unit, the heat transfer that is needed to heat the outdoor air until the desired supply temperature is [33]:

$$q_S = \dot{V}_e \cdot \rho_{air} \cdot C_{p_{air}} \cdot (T_s - T_e) \text{ [kW]} \tag{15}$$

The extra heat transfer required to warm the supply air in the reheating coil after this one be pre-warmed passing in the preheating coil is given by [33]:

$$q_{hc} = \dot{V}_e \cdot \rho_{air} \cdot C_{p_{air}} \cdot (T_c - T_{rec}) \text{ [kW]} \tag{16}$$

If the cooling coil is not working, what happens during the winter season, the supply air temperature will be the same as the air temperature after the heating coil, i.e. Tc=Ts.

The actual heat transfer in the heat recovery unit is the product of the temperature ratio η_T (calculated with equation 11) and the theoretical maximum heat transfer determined by equation 14:

$$q_{actual} = \eta_T \cdot q_{max} \text{ [kW]} \tag{17}$$

This value represents the heat saved in the supply air stream and at the same time the recovered heat in the exhaust air stream.

3.1.3. Pump

There are different pumps working along the HVAC system positioned in units where there is circulation of liquid fluids, as for example chillers and boilers. Nowadays is more frequent to find pumps working with controlled flow instead of continuous (with constant-speed pumps). That control is possible thanks to several other mechanisms possible to join to the pump, like bypass valves to reduce or balance the flow.

The overall efficiency of a pump used in a HVAC system depends on several factors as the efficiency of the pump and motor, the efficiency of the pump control and how well technicians maintain the pump and its related components [53]. The hydraulic efficiency of a pump is the relationship between the power the pump imparts to the water compared to the power input from the motor that can be calculated by [54]:

$$\eta_p = \frac{\rho \cdot g \cdot \dot{V} \cdot H}{Power} \tag{18}$$

Where ρ represents the fluid density (kg/m^3), g the acceleration due to gravity (m/s^2), $\dot{V}$ the flow (m^3/s) and H the head (m).

Each pump has a pump characteristic curve associated to it that considers the flow and the differential pressure values. That curve joint to the curve representing the system characteristics is essential for a correct pump choice. The duty point is the intersection between the pump performance and the system's characteristics curves.

Nowadays, with easy access to simulation software is possible to observe important characteristics as the duty point and observe how the pump is working. For the present HVAC system in building 99, the pump characteristics of the pump insert in the heat recovery units (run-around heat coil with ethylene glycol 25% concentration solution), were simulated on-line with the tool provided by the manufacturer.

3.1.4. Economic savings

As previously indicated, one motivational aspect to include heat recovery units in HVAC systems is the economic savings connected to the decrease of needed energy to heat the outdoor air until the desired supply air temperature. The annual heat loss of one heat recovery unit can be expressed by the next equation:

$$Recovery = (1 - \eta_T) \cdot \dot{V}e \cdot \rho_{air} \cdot C_p \cdot deg.hours \text{ [kWh]} \tag{19}$$

Where η_T is the efficiency of the heat recovery unit, $\dot{V}e$ the supply air flow volume (m^3/s), ρ the density of the supply air flow (kg/m^3), C_p the specific heat of the air (kJ/kg.ºC) and *deg.hours* the degree hours for the city (h.ºC). The degree hour parameter is commonly used to estimate the energy requirements for heating or cooling since it represents the number of degrees by which the hourly average indoor temperature is above or below a standard temperature. The total degree hours per year can be find in a specific table considering the place where the system under analysis is located and the desired temperature as also the annual average outdoor temperature of the same place.

3.2. Description of the design

The problem stated to analyse makes reference of two separated ventilation systems that supply the air needs for one single building, providing each one an amount of air for the total required. Since they work together to supply a total flow, in order to simplify the initial design, two similar air-handling units were installed in the underground level of the building. Equally, on the attic/roof there are another two exhaust air units. In this section, it is presented the design of both systems as well as their characteristics.

3.2.1. Description of the ventilation system in the building

The University of Gävle was founded in 1977 with inauguration of the new campus in 1996. In the subsequent years, some departments and divisions were added to the previous construction made by the reconstruction of regiment barracks. Nowadays, it consists of 18 different buildings, some of them interconnected via underground too. Building 99 (also named "Freja") is one of them, since it is directly connected to building 12 that acts like a bridge to main buildings. The 1st floor (or the basement) has several bathrooms, open spaces, small rooms and theatres, multimedia rooms, a kitchen and two fan rooms containing the supply air-handling units. The building has more than 100 different rooms spread over six floors, being most part of them classrooms of different sizes. There are also some laboratories of physics and other sciences in the 4th and 5th floors. In the 6th floor are located, among others, some researcher's offices and two connections for the rooms in the attic that have the exhaust air equipment.

Each room or open space in this building (similarly to other buildings at the university) has its own requirement of airflow rate, accordingly to the area of the space, the type of activities going on and the heating and cooling needs. Some rooms as the ones where is expected to have sudden changes in the number of people inside (like common lecture rooms, conference rooms, passages and waiting areas) or in some rooms that have to verify constant environmental conditions despite possible increase in the number of people (like lab rooms), have the possibility to increase the airflow rate from a base value to a new value in accordance with the needs. This change is made automatically through a signal of sensors (e.g. temperature, CO_2) to the controlling software or manually by adjustment of the setting button in the room (when it exists), like a VAV system. The dampers regulate the amount of air flowing through the ducts. Some of the rooms in the building have more than one diffuser.

As it was referred, Building 99 has two separated air-handling units for its mechanical ventilation system: LB20 and LB21. Ventilation system was built after the reconstruction of the building, only in the year of 2000.

The running time of the ventilation system is 24 hours per day, seven days a week. This happens because this building contains laboratory rooms with constant experiments going on that require the ventilation working.

Since the temperature outside strongly influences the indoor need of fresh air, the supply temperature changes depending on a curve based on the outside temperature. With -20ºC outside, the supply air temperature is 19ºC and with + 20ºC, it changes to 17ºC.

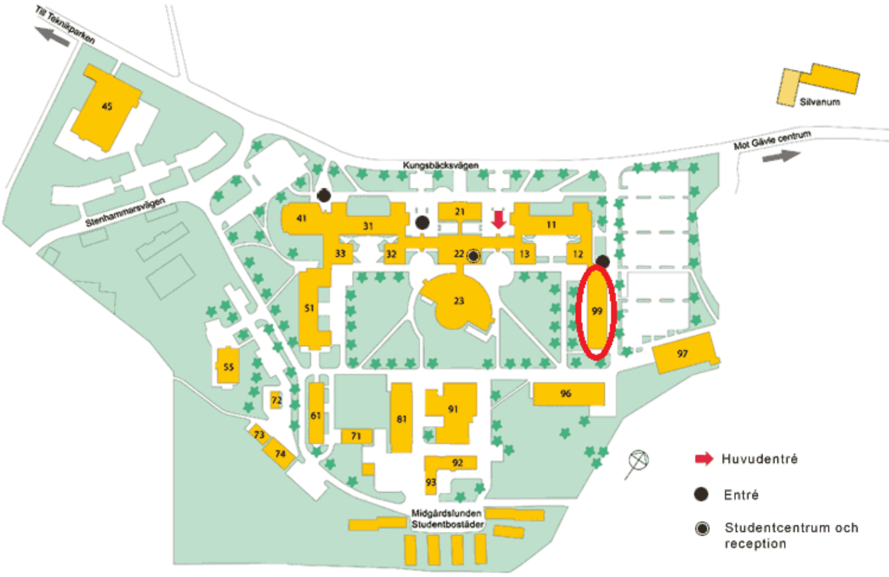


Figure 17: Map of University of Gävle Campus with building 99 highlighted [55]

3.2.2. Characteristics of the air-handling units

The air-handling units in building 99 were both developed by NOVENCO and belong to the category Climaster ZP-135. This model has capacity to supply air within 6 and 18 m^3/s. Their casings consist of hot-dip galvanized steel sections and alu-zinc steel sheets separated by an insulation layer of 50 mm mineral wool and were assembled into complete units on site. In wet sections, as cooling coils and humidifiers, there are watertight drip pans with drains to prevent accumulation of water inside the components.

The shut-off dampers in the beginning of the unit in the basement and in the roof are used to prevent the incoming air to enter when the unit is out of operation in contrast to the dampers after the fan in the air-handling unit whose primary functions are essentially to regulate the airflow.

The model ZP-135 has 12 filter units per air path.

The first heating coil (the one part of the run-around heat exchanger) is used to preheat the outdoor air and has 6-15 tube rows. The second heating coil (or reheating coil) has 1-5 tube rows and has a temperature sensor that transmits signal to the operator and to the control unit (TC). In these ventilation systems, both heating coils are supported by water coming from the district heating system in the city of Gävle. After other inspection section, a cooling coil with 2-10 tube rows is used for cooling and dehumidifying (when necessary) the outdoor air.

Each ventilation system have two axial fans: one inside the air-handling unit in the basement for the incoming outdoor air and other in the attic in the exhaust air-handling unit for the outgoing air. In the supply air-handling unit, there are axial flow fans with adjustable blades and a short diffuser, with a rotor diameter of 900 mm and hub diameter of 403mm. The fans are fixed on rubber anti-vibration mountings and each fan has an inspection door with an inspection window. Outside of each door, there is a manometer for filter guard. In the supply air-handling unit, each fan has the maximum capacity to work at 2279 rpm and an optimum efficiency above 80% when working with blades with angles precisely calculated. The load of the fan is adjusted periodically according to the need of air in the rooms, increasing when is necessary a general increase of airflow in the zones. Coupled to the fan, it is also possible to find a frequency converter.

Individual components as filters, humidifiers or silencers are chosen taking into account the air velocity in the front area in the component. In the following figure is presented a scheme of the ventilation system for LB20 or LB21. In its legend are identified the components over the supply air-handling and extract unit and between brackets it is possible to see the code of each one of these components in order to search for technical details, length, weight and function in the catalogue of the producer’s handbook [56].

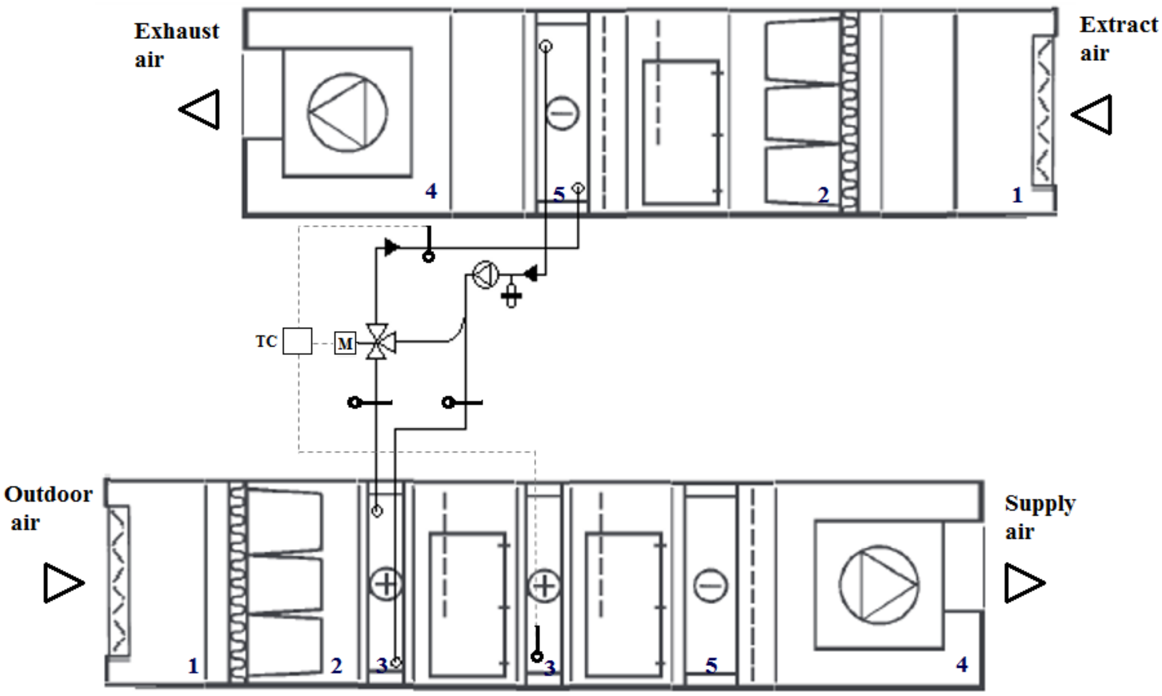


Figure 18: Scheme of LB20 or LB21. 1- Shut-off damper (SF); 2- Pocket type fine filter (FF); 3- Heating coil (LB); 4- Axial flow fan (AC); 5- Cooling coil (QA)

Each heat and cooling coil has water flowing through it. It is very important to ensure that the water is flowing with a constant rate. To guarantee that, there is a constant control of the water temperature by means of a separate circuit in connection with a shunt-arrangement close to the coil as shown in Figure 19.

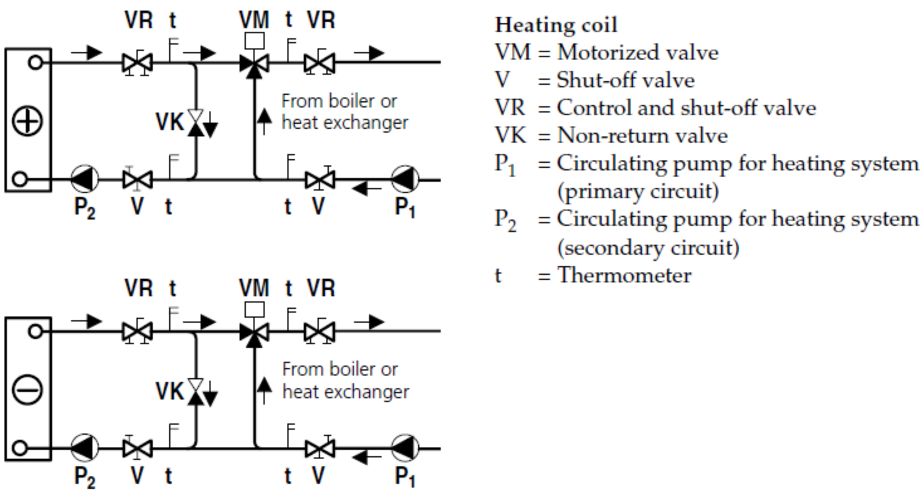


Figure 19: Controlling system to ensure constant water flow through the heating/cooling coil [56]

The airflow protocol elaborated when HVAC system was installed, discriminates the airflow rate for each air diffuser in different rooms all over the building and presents post installation testing values according to different measurement methods: measuring pressures plus use of a diagram or measuring pressures plus k-factor. The projected airflow rates are different according to the unit and also differ from a maximum and a minimum value. The minimum projected supply airflow rates passing through the supply air-handling units are 10 m^3/s for both LB20 and LB21, while the maximum values are 11,4 m^3/s and 12,9 m^3/s, respectively. In LB20, the minimum exhaust air flow is 7,55 m^3/s while in LB21 is 4,64 m^3/s. The maximum values are 11,16 m^3/s for LB20 and 11,60 m^3/s for LB21.

As previously referred, there are some sensors (data loggers) placed both in supply and exhaust air units that transmit temperature and pressure values to a controlling software. These sensors are insert in the system and they are identified as *GP* or *GT* plus an identification number.

3.3. Measurements

Despite the already existent sensors, to calculate the efficiency of the heat recovery unit, are necessary additional measurement points of different air temperatures. Due to the inexistence of openings to insert the hot wire anemometer in the air-handling unit, these ones were subsequently made with a drill and then closed with a rubber itself for the purpose. Those temperatures were measured in four different places of the already existing sensors: for the outdoor incoming air before the heat recovery unit, in the basement (T_e); for the same string of air but now after the heat recovery unit in the basement (T_{rec}); for the extract air before the heat recovery unit in the attic (T_{ext}) and after the same coil in the roof (T_{exh}). Together with those temperature measurements, the velocity was also taken for the exhaust air (v_{exh}), for the outdoor air (v_e) and for the preheated air after its passage by the heat recovery unit (v_{rec}), that is supposed to be similar. Before the damper, in the entrance of the air-handling unit, four holes were made equally spaced in the vertical, in order to get a more genuine value (with the average of them).

Considering the fluctuations given by the anemometer when measuring especially air velocities, it was measured more than one time the same place, being considered the final value, the average of the measurements. The same happened for water measurements of flow, temperature and pressure difference.

The following figure is a simple scheme of the existing ventilation system for LB20 or LB21 and has represented with a red point the places where temperatures and velocities of the air were measured.

In appendices B, B.1 and B.2 is possible to observe more detailed schemes of the whole system, including the complete design of the heat recovery unit, being B.1 and B.2 print screens of the system working in the day 09/12/2015, winter period. In those two figures, are represented the existing sensors in the different points of the system showing punctual temperature and pressure values on them and also the fan load both in the entrance and in the outlet of the air-handling units.

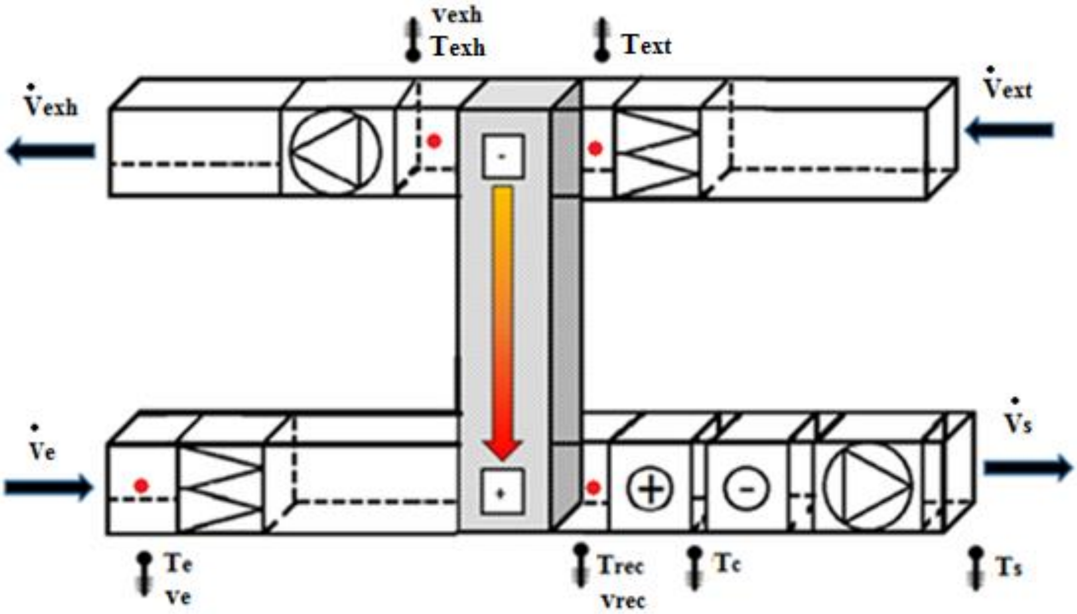


Figure 20: Simple scheme of the ventilation system with red points representing the locals of temperature and velocity data acquisition

3.4. Measurement devices

In order to measure the different values of temperature, pressure and flow at different points of the system, two instruments are used, being one of them directed to air flow and other to water flow. The periodical measurement of certain quantities is fundamental to evaluate how the system is working and make adjustments (if necessary), controlling its performance and ensure a correct operation.

The measurement of temperature, pressure and flow in the recovery unit are performed with TA SCOPE balancing instrument. This instrument is made for hydronic systems, being very accurate and easy to use, with interactive software that ensures fast commissioning process and possibility of wireless communication between the two components: *Handheld Unit* and *Differential Pressure Sensor Unit* (shown in figure 20). The first one is a computer-based unit programmed with the TA valve characteristics and the second a differential pressure (ΔP) sensor that communicate wirelessly with the handled unit, having also the possibility to be connected via cable [57].

Figure 21: TA-SCOPE [57]

The range for temperatures measurement is from - 20 to 120°C. The following table shows the possible measurement deviations by the instrument, presented in the handbook of it.

Table 5: Some difference values (error scale) of the liquid measuring instrument [57]

Differential pressure	**TA-SCOPE**	0.1 kPa or 1% of the reading, whichever is the highest
	TA-SCOPE HP	0.2 kPa or 2% of the reading, whichever is the highest
Flow		As for differential pressure + valve deviation
Temperature		< 0.2 °C

To get the instantaneous values of air velocity and temperature, an anemometer is used. An anemometer is a device that measures the direction and the velocity of a fluid, in this case air. There are several types of anemometers that can be used. In this experiment, it is used a thermal anemometer, more specifically a hot wire anemometer, which technique depends on the convective heat loss to the surrounding fluid from an electrically heated sensing element, admitting that only the fluid velocity varies. Therefore, admitting this, is possible to get the value of heat loss that will reveal the velocity of the stream.

This type of anemometer is the most indicated device to use in this HVAC systems since its operating principle is simple and, more important than that, because it has a small probe that can be easily insert in narrow places on the air ducts. In order to collect more realistic values of velocity, the sensor have to be placed normally to the flow and preferably in a zone where it has laminar pattern. With this type of device, it is also possible to get the temperature value of the flow. On the following picture is possible to see an example of a hot wire anemometer. The errors associated to this instrument are ±3% for velocity measurements, ±0,3°C for temperature, ±3% for relative humidity and ± 1 Pa for pressure difference.

Picture 1: Anemometer used

3.5. Credibility and trustworthiness of results

In order to get accurate results, several aspects were taken into consideration. Each value was only recorded after a waiting period of several seconds until it has reached stability. In cases where was difficult to observe constancy in the values, more samples were taken to consider later the average (among a bigger range of values).

To confirm that the temperatures observed in the analogue thermometers in the preheating coil were precise, they were later checked with a thermal sensor insert in the cavity were the analogue thermometers are. The results were only different in some decimal places since in the analogue thermometers the results have zero decimal places.

Since the objective of this study is to analyse how the heat recovery unit is working in a cold day, it was not necessary to install data loggers in order to get the supply and exhaust airflow values as also air temperature values in the supply, extract, exhaust and after the preheating coil.

As previously mentioned, in the beginning of this study, the supply airflow was calculated with the values of velocity measured in the entrance of the air handling unit, right before the shut-off damper. Due to large fluctuations in the air velocity expressed in the anemometer (since that is a turbulence zone), it was then done another acquisition of air velocity right after the preheating coil since this zone presented itself as a zone with more organized flow and less fluctuations. In order to confirm that the value of airflow after the preheating coil was more accurate than the one measured in the entrance of the air-handling unit, other calculations were made. For that, 3 different supply airflow were considered: case 1 – The supply airflow was the average between the airflow in the entrance of the AHU and after the preheating coil; case 2 – The supply airflow was the one measured after the preheating coil; case 3 – The supply airflow was the one measured in the entrance of the AHU. Then, for all cases, it was calculated the heat transfer from the extract air to the heat recovery unit as also the heat transfer from the heat recovery unit to the supply air.

The lower difference between them would represent the best case since the exhaust airflow was constant. With that condition, case 2 was found to be the best assumption for the supply airflow. However, the previous condition was not the only one taken into account. To confirm that case 2 represented the best airflow, it was also made a comparison between design airflow (provided on the airflow protocol) and measured airflow for the same 3 cases. However, considering this factor, case 3 was indicate as the best to consider since the difference between the projected and the measured airflow were smaller. Yet, case 3 was excluded after calculation of the effectiveness of both heat recovery units, since their values were above the common values of effectiveness for this type of system.

Chapter 4 – Results and analysis

This chapter contains the results obtained by the measurements made in LB20 and LB21 in the University of Gävle. As previously introduced, the main objective of data gathering was to calculate the efficiency of the heat recovery units that is directly linked with energy savings. A complete (air and water) data collection for the whole system was performed in a winter day, when was supposed to have a higher load, and then another punctual acquisition of air temperature and air flow values were taken in a warmer day. Energy savings associated to the energy transfer rate were calculated as also the required heat without the use of any heat recovery unit to pre-heat the outdoor air and the extra heat transfer supplied from the district heating water in the reheating coil.

4.1. Flow and temperature

Air

In both days the values of supply and exhaust airflow were measured to understand the impact of the supply air volume in the heat recovery unit and to analyse the balance of the system by getting the ratio between the supply and the exhaust airflows. It is common to find mechanical ventilation systems with heat recovery that are slightly under-balance, with supply airflow rate usually lower than the exhaust airflow rate [58]. Tables 6 and 7 show the obtained values for the supply and exhaust airflow rates in LB20 and LB21, respectively. In general, it is possible to observe that the exhaust airflow is larger than the supply airflow turning the ventilation system under-balanced. However, these values are very similar to what is considered a balanced system with a balance ratio of one unit. It is also notable that both supply and exhaust airflows are larger in LB21 when comparing it with LB20, indicating that one air-handling unit is working with higher load that the other, supplying and removing more air to and from the building. The values for the supply and exhaust air flow volume were calculated considering the velocity measured with the hot wire anemometer in the section after the pre-heating coil and the cross section area of the enclosure, multiplying with each other.

Table 6: Airflow measurements for LB20

LB20	10/02/2016	21/04/2016
Supply, $\dot{V}e$ (m^3/s)	5,50	6,18
Exhaust, $\dot{V}exh$ (m^3/s)	6,53	6,55
Ratio	*0,84*	*0,94*

Table 7: Airflow measurements for LB21

LB21	10/02/2016	21/04/2016
Supply, $\dot{V}e$ (m^3/s)	7,41	8,23
Exhaust, $\dot{V}exh$ (m^3/s)	7,11	9,05
Ratio	*1,04*	*0,91*

As previously referred, in both units the supply airflow was firstly measured in the entrance of the air-handling unit, along 4 vertical openings done right after the casing of the shut-off damper. Due to large fluctuations in the velocity values of the very turbulent air in that region, it was measured also the air velocity after the heat recovery unit (or preheating coil) in the supply air-handling unit. Despite also here the values of the velocity of air were somehow not constant, the airflow taken there was considered more accurate, as previously mentioned in section 3.5.

Temperature is one of the most important variables to be considered in HVAC systems, especially when the component in assessment is the heat recovery unit. As mentioned above in methodology, the temperature values of air, water and water-glycol solutions were taken for further analyse. In tables 8 and 9 are recorded the temperature values of airflow in relevant points in the air-handling units (shown in the next scheme), i.e., outdoor air temperature and air temperature after the reheating coil in the supply AHU and extract and exhaust air temperature in the exhaust AHU. Since during these measurements the cooling coil was not working, it was admitted that the temperature after the reheating coil was equal to the supply temperature, that between 0°C and 10°C is considered to be approximately 18°C.

Table 8: Air temperature measurements for LB20

LB20	10/02/2016	21/04/2016
Outdoor, Te (°C)	1,45	9,68
After preheating coil, Trec (°C)	9,15	14,60
Extract, Text (°C)	19,95	20,90
Exhaust, Texh (°C)	11,50	15,78

Table 9: Air temperature measurements for LB21

LB21	10/02/2016	21/04/2016
Outdoor, Te (°C)	1,50	10,95
After preheating coil, Trec (°C)	9,66	15,18
Extract, Text (°C)	19,55	20,70
Exhaust, Texh (°C)	11,00	16,38

The main difference that is possible to see in the previous tables is the one concerning to outdoor temperatures between the first (10/02/2016) and the second day (21/04/2016) of measurements, i.e. a change from ~1.5°C to ~10°C. This big difference is the basis for the explanation of differences of the temperature ratio in the heat recovery units. Is easy to observe that the extract air temperature has similar values (~ 20°C) no matter what is the outdoor temperature, being that one higher than the supply air temperature due to the heat gains in each room, as it was expected to observe since most part of the rooms are occupied. Despite very similar, it is possible to see that in april the extract air temperature is slightly higher than in February probably due to the solar heat gains by the building that has a large number of windows or glass walls. The same does not happen with the exhaust air temperature. Since the outdoor air temperature is higher in the April, it is not necessary to preheat so much the supply airflow when passing through the preheating coil, what is possible to see in the higher values of the exhaust air temperature both in LB20 and LB21, when comparing with the measurements in February. In the first day of measurements (February), the exhaust air temperature was lower than the air temperature of the extracted air due to the high heat transfer from the run-around coil. A general small difference between temperature values in LB20 and LB21 can be explained by the fact that the measurements were not made in the same exact moment of the day but yes with some time between each one, with the day warming up as the time passed.

In order to visualize better the previous mentioned points and the ones that will be cited later concerning to the glycol-water solution and the secondary water cycle, the following figure (also shown in appendix B) should be used in parallel with the analysis. The outdoor air starts its journey by passing through the damper A1, being filtered and heated until the desired supply temperature (in the figure the last point before the air enters into the zones). The points where the temperature of the air streams was taken are shown in red. After the supply the heat to the zones, it is extracted, filtered and cooled before be exhausted by the fan F2. All the points where there is transfer of heat from a liquid medium for the air streams are represented in blue with letters H, S and C, depending if the air is being preheated, reheated or cooled.

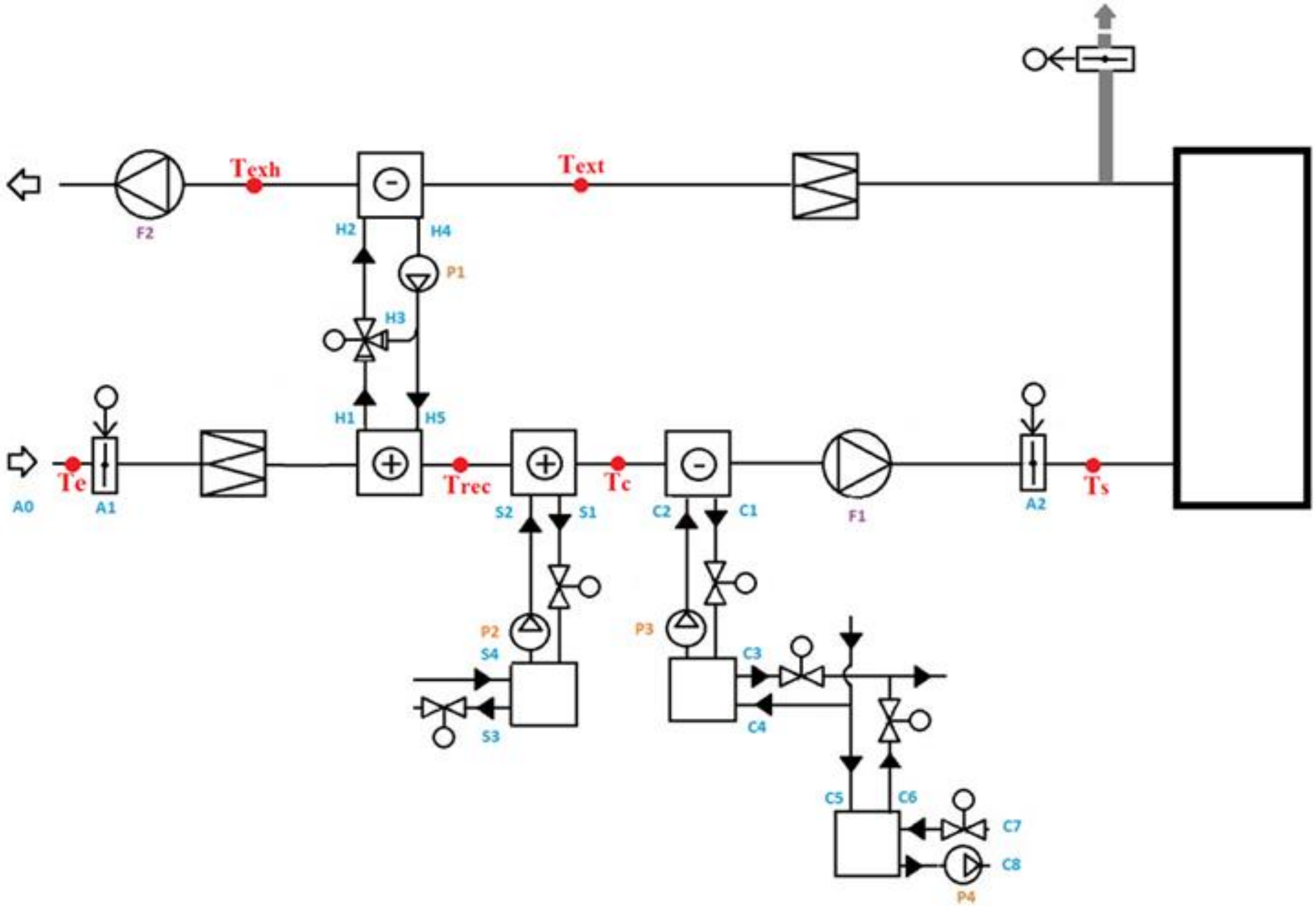


Figure 22: Scheme of the Ventilation System for LB20 or LB21 with important points (T, H, S)

Glycol-water solution

The measurements taking place over the heat recovery unit were performed on STAF balancing valves 80, in the sections H3 and H4 in the system design. H3 represents the point right after the balance valve through where is possible to bypass the heat transfer medium and H4 represents the point and the balancing valve before the pump where the heat transfer fluid passes after the cooling coil in the rooftop unit. The temperature values of H1 and H5, were presented in analogue thermometers in the preheating coil casing, representing the temperature after and before the working fluid passes through it, respectively. The following tables show the average values of the measurements done in the first day for LB20 and LB21. In both valves, the TA-SCOPE measurement device were constantly alerting for errors in the system turning very unstable the data acquisition.

Table 10: Average measurements of flow in the heat recovery unit of LB20, 10/02/2016

LB20	Flow (m^3/s)	ΔP (kPa)	T (°C)
H1	-	-	8
H3	2,35	5,34	15,50
H4	10,11	64,57	15,49
H5	-	-	15

Table 11: Average measurements of flow in the heat recovery unit of LB21, 10/02/2016

LB21	Flow (m^3/s)	ΔP (kPa)	T (°C)
H1	-	-	8
H3	4,65	18,67	14,67
H4	7,39	16,66	14,63
H5	-	-	14

When it concerns to the flow and pressure difference in the heat recovery unit, there are considerable differences between LB20 and LB21. The bypassed flow in the circuit (H3) of LB21 is twice the value in LB20, despite the flow after the cooling coil (H4) in the rooftop unit be higher in LB20. Both in LB20 and LB21 the temperatures before and after the working fluid exchange heat with the supply airflow are very similar and about 15°C. In tables 8 and 9 it is possible to observe that the air temperature after the heat recovery unit was around 9-10°C in this day. It is also notable the difference in pressure difference in H4 between LB20 and LB21, being significantly smaller in LB21. The pressure difference in H4 for LB21 has a very unrealistic value since it should be even higher than the value of 65 kPa observed in LB20. The high level of fouling in the system is one possible cause for that. Since H4 is located in the basement instead of the attic, it is not possible to know if the loss of heat is significant across the pipes along 25 meters of height.

Secondary water in the reheating coil

The reheating coil has a very important function in ventilation systems placed in countries with cold climates like Sweden. In the case of building 99 at University of Gävle, the reheating coil has as working fluid hot water coming from the local district heating. As shown in appendix B, this component has two control and shut-off valves where is possible to get the values of temperature, pressure difference and flow with the TA-SCOPE instrument. The following table shows the values of those variables in the water cycle after the passage of it in the reheating coil (S1) and also in the second shut-off valve when the water has no more heat to transfer (S3) and is returned to the cooled water system to be reheated again in the district heating plant.

Table 12: Measurements of the water circuit in LB20 reheating coil, 10/02/2016

LB20	Flow (l/s)	ΔP (kPa)	T (°C)
S1	3,16	39,64	20,21
S3	0,646	0,638	19,77

Table 13: Measurements of the water circuit in LB21 reheating coil, 10/02/2016

LB21	Flow (l/s)	ΔP (kPa)	T (°C)
S1	3,05	30,46	21,53
S3	0,834	0,913	20,53

The values are similar in some way for both systems since they have the same amount of hot water coming from the grid until the control unit of the reheating coil (illustrated in figure 19), of about 3.8 l/s. LB21 has less water flowing through the reheating coil and more water being bypassed and is possible to see that the water temperature after the reheating coil is approximately 1°C more than in LB20. Is easy to observe that the pressure difference is higher when the flow is higher.

As previously mentioned, the reheating coil has one control unit to ensure a constant value of water flow through the pipes. All the points *S*, related to the reheating coil, have their temperatures shown in analogue thermometers at the same unit. The temperature of the supply water coming from the district heating (S4), as also the temperature of the water that supplies the reheating coil (S2) were possible to see on those thermometers. Their values for LB20 and LB21 were as follow:

Table 14: Water temperatures in the reheating coil for both LB20 and LB21, 10/02/2016

	LB20	**LB21**
	T (°C)	T (°C)
S2	22,50	23,00
S4	41,00	26,50

Here the big difference is seen in the temperature of the supply water (S4) between both systems. While the supply temperature in LB20 is 41°C, in LB21 is only 26,5°C. This difference can have different possible explanations: the first is that the analogue thermometer can be damaged, showing a different value from the real (in the system was possible to observe that this one had been already replaced) and the second is that the value is correct and the difference of temperatures between in and out water flows is not important to be big as long as all the necessary heat is transferred effectively.

Air flow protocol

The following tables show the projected supply and exhaust airflows for LB20 and LB21 upon the design in 2000. Different values of minimum and maximum airflow are shown once the HVAC system in building 99 has working principle of VAV-system with adjustable airflows according to the needs in different rooms. The minimum values take into account the minimum projected airflows for the rooms were is possible to change the airflow and constant airflow for all the other rooms and the maximum airflow takes into account the maximum demand in the rooms where is possible to change the airflow while all the others continue with constant airflow.

Table 15: Projected airflow for LB20

LB20	Units	Minimum	Maximum
Supply	m^3/s	9,92	11,36
Exhaust	m^3/s	7,55	11,16

Table 16: Projected airflow for LB21

LB21	Units	Minimum	Maximum
Supply	m^3/s	9,79	12,84
Exhaust	m^3/s	4,64	11,59

The projected minimum supply airflow is about 10 m^3/s for both units. It is easily seen that the minimum exhaust airflows are different, being smaller in LB21 than LB20. This difference is because the ventilation in building 99 is not divided in two very equal systems with different needs in different rooms placed all over the building. The fact that measured values of supply and exhaust air flows are different from the projected values can be explained by several different possible reasons. The first is that when ventilation systems are projected, the airflow values are chosen for a completely cleaned system, admitting no dirty in the fans, in the air-handling unit or even in the air ducts. Second is the pressure drop verified along the system. Other possible explanation could be that the air flow values were collected in the more or less turbulent air. After some years of operation, it is expected some changes in the system, justifying this difference between measured and projected values. It is also important to point out once again that the measured values are punctual data acquisition, not representing any average value for the day/month/year.

4.2. Efficiency

In this analysis, all the calculated values are for dry heat recovery efficiency of the heat recovery units since this type of heat recovery device has no transference of latent heat as it is possible to find in others like the rotary wheel. Table 17 directed to LB20 and Table 18 to LB21, comprise the variables that influence the efficiency calculation. In both cases, the supply airflow considered is the one that uses the velocity measured after the preheating coil in the AHU.

Table 17: Efficiency of heat recovery unit in LB20

	10/02/2016	21/04/2016
Outdoor air temperature, Te (°C)	1,45	9,68
Temperature after the heat recovery unit, Trec (°C)	9,15	14,60
Extract air temperature, Text (°C)	19,95	20,93
Supply airflow, $\dot{V}e$ (m^3/s)	5,500	6,180
Exhaust airflow, $\dot{V}exh$ (m^3/s)	6,532	6,550
Ratio	*0,842*	*0,943*
R	*1,000*	*1,000*
Temperature ratio, η_T	**0,350**	**0,413**
Sensible effectiveness, ε	**0,416**	**0,438**

Table 18: Efficiency of the heat recovery unit in LB21

	10/02/2016	21/04/2016
Outdoor air temperature, Te (°C)	1,50	10,95
Temperature after the heat recovery unit, Trec (°C)	9,66	15,18
Extract air temperature, Text (°C)	19,55	20,70
Supply airflow, $\dot{V}e$ (m^3/s)	7,408	8,230
Exhaust airflow, $\dot{V}exh$ (m^3/s)	7,108	9,048
Ratio	*1,042*	*0,910*
R	*1,042*	*1,000*
Temperature ratio, η_T	**0,471**	**0,394**
Sensible effectiveness, ε	**0,471**	**0,433**

The quantity above mentioned as *Ratio* in tables 6 and 7 and also in the previous two, is the real ratio between the supply and the exhaust airflows. The multiplying factor in equation 12, named as *R*, represents the ratio between the supply airflow and the minimum airflow value between supply and exhaust, which is used to calculate the sensible effectiveness, whereas *Ratio* is used to calculate the temperature ratio.

The value that claims more attention is the low temperature ratio in LB20, of 35%, when comparing the same parameter with LB21, of 47%, for very similar outdoor conditions. Since the ratio between airstreams is about 0.8, this leads to this very low value of temperature ratio, being possibly below its real value. In general, is possible to see that the sensible effectiveness is higher than the temperature ratio in LB20 since the multiplying factor *R* is higher than the *Ratio.*

In the second day of measurements, with higher outdoor temperature, both units perform quite similarly within each other. When comparing how well the heat recovery unit works in LB20 and LB21, is possible to observe that in the case of LB21 the efficiency is higher with cold outdoor climate what is expected but not possible to observe in LB20.

Regarding to the efficiency definition for heat recovery units given by ASHRAE, is possible to see that only for LB21 in the first day of measurements the sensible effectiveness is equal to the temperature ratio. This can be explained due to the unbalance between air streams with *R* is higher than 1, once the supply airflow is higher than the exhaust airflow. In all the other cases the exhaust airflow was bigger than the supply airflow.

Since both sensible effectiveness and temperature ratio can be used to calculate the efficiency of a heat recovery, for the heat transfer and economic savings calculations, the temperature ratio will be considered since this one represents a more realistic value.

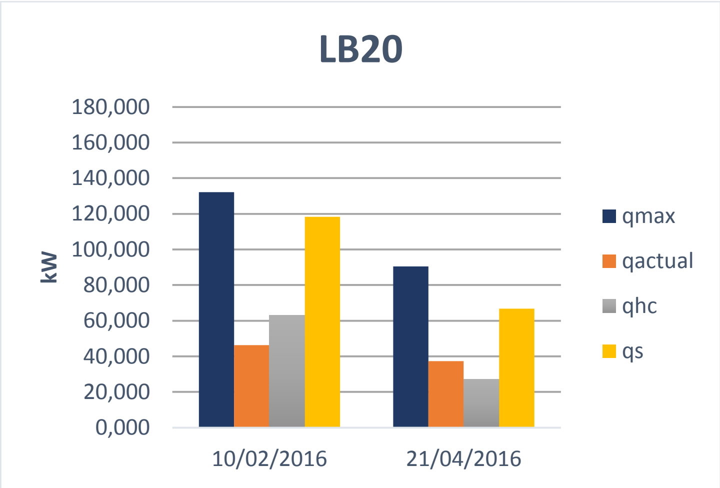
4.3. Heat transfer

To understand the impact that the heat recovery unit has in the HVAC system, different values of heat transfer associated to the system were calculated. Among them is the heat transfer that is needed to heat the outdoor air until the desired supply temperature (qs), the theoretical maximum possible heat transfer by the heat recover unit (q_{max}), the actual heat transfer associated to the heat recovery unit (q_{actual}) and the extra heat transfer by the reheating coil that is needed to rise the temperature of the outdoor air until the desired value (q_{hc}). Those quantities were calculated considering the equations 14, 15, 16 and 17 and are possible to see in appendices F1 and F2.

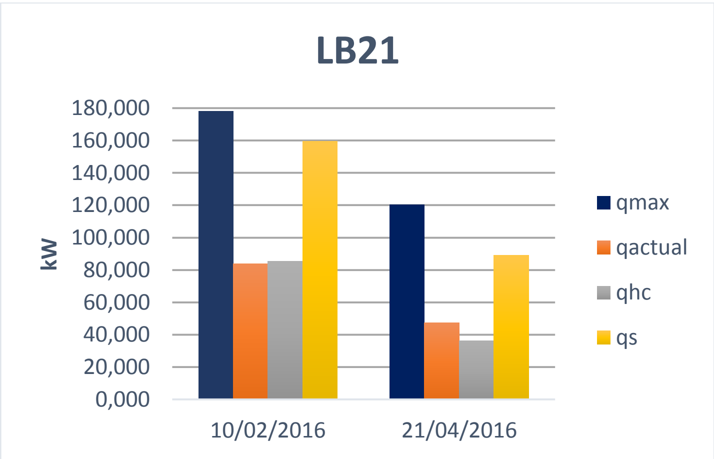
In Graph 1 and Graph 2, it is possible to observe the same tendencies of heat transfer between the units of LB20 and LB21. In the first day (10/02/2016) the heat transfer required to heat the supply air without this one be pre-heated in the heat recovery unit, q_S, is almost the double of the one in second day. This discrepancy is due to an outdoor difference of sensibly 8°C between those days. When comparing the actual heat transfer of the heat recovery unit in February (considering the measured temperature ratio), q_{actual}, is notable that it fulfils a significant portion of the total required heat needed to heat the supply airflow in LB20 (39%) and at least half of the total required heat in LB21 (52%). Due to the big difference between the outdoor temperature (1.5°C) and the supply temperature (18°C) in February, the extra heat transfer that also has to be transferred by the reheating coil, q_{hc}, is bigger than the heat transfer "saved" by the heat recovery unit, q_{actual}. In April (the warmer day), since the outdoor temperature was sensibly 9°C (being more similar to the desired supply air temperature), it was possible to see that the heat transfer from the reheating coil, q_{hc}, was not so high as in the first day, being even smaller than the heat transfer associated to the heat recovery unit, q_{actual}.

This pattern is also seen in LB21, however with higher intensities due to large supply airflow. Is also possible to observe that the actual heat transfer by the heat recovery unit, q_{actual}, is almost the same as the extra heat transfer from the reheating coil, q_{hc}. The difference between these two quantities is higher in LB20. In February, when there is need to have higher q_{actual}, there is a difference of heat transfer of about 38 kW between LB20 and LB21, being the savings higher in LB21.

If the heat recovery unit had an effectiveness of 100%, it would not be necessary any extra heat from the reheating coil since q_{max} is bigger than q_S for both days and units.



Graph 1: Heat transfer in LB20 on both days of data acquisition



Graph 2: Heat transfer in LB21 on both days of data acquisition

4.4. Pump

The simulation of the operating conditions of the pumps in the run-around heat recovery units were made in Grundfos website for the pump type TP 50-180 / 2. The TP pumps are used in commercial buildings for air conditioning and heating applications and are electro-coated to ensure high corrosion resistance. In this type of pump, the pump and the motor are separate units. Both simulations for LB20 and LB21 considered a 25% ethylene glycol solution with temperatures around 15-16°C, for the corresponding flow values in the points H4, i.e., in the returning pipe from the cooling coil in the exhaust AHU. The existence of a pump is fundamental to ensure that the heat transfer fluid continues to circulate, transferring heat from the extract air to the supply air.

In appendix D1 it is possible to see the nominal duty point of the two similar pumps, considering a static head of 25 meters. The point of maximum efficiency (100%) takes places when the flow is 17.1 m^3/h and the head 10.2 m.

After the insertion of the operating conditions verified in the first day of measurements, it is possible to observe (in appendix D2) both the operating point and the duty point of the pumps for LB20 and LB21. In the same appendix are presented three different cases. The first chart counting from the left to the right represents LB20 pump. It is possible to see here a pump efficiency around 65% for the operating point with flow of 10.1 m^3/h and pressure difference of 64.6 kPa. When it concerns to LB21 pump, two different figures are shown with different pressure difference between them. In the middle figure of appendix D2, the insert values of flow and pressure difference are the ones shown in table 11. With those values, it is seen that in LB21 a) is not possible to get any representation of the operating conditions of the pump. In the right chart, LB21 b), the flow was maintained with the value of 7.39 m^3/h but the pressure difference was changed from the measured 16.7 kPa to 46.8 kPa, being this last value an estimation of a more “real” value. With this last assumption, the efficiency of the pump in the operating conditions is around 47%.

With minimum efficiency of 70%, acceleration of gravity of 9.81 m/s^2, 25 m of static head, density of 1039 kg/m^3 and the correspondent flow for each system, is conceivable to determine theoretically the power that the pump requires manipulating equation 18. The results of that calculation are presented in table 19. Is possible to see that due to a larger flow in LB20 when comparing with LB21, the power of the pump is also larger.

Table 19: Power of the pump in the heat recovery unit in LB20 and LB21 when the efficiency of the pump is 70%

	Power [W]
LB20	1023
LB21	747

If instead of assuming a minimum efficiency of 70% consider the efficiency presented in the charts of appendix D2 where the operating efficiency resulted from the simulation is 65% for LB20 and 47% for LB21, the new power consumption of the pumps insert in the run-around coils are:

Table 20: Power of the pump in the heat recovery unit with efficiencies of 65% for LB20 and 47% for LB21

	Power [W]
LB20	1101
LB21	1113

Looking to table 20 it is seen that despite LB21 have a lower pump efficiency when compared with LB20, the pump consumption is very similar and around 1.1 kW. This fact is possible since the flow of the ethylene-glycol solution in LB21 is lower than in LB20.

When comparing the pump consumption calculated admitting the minimum efficiency of 70%, the power is relatively smaller than when calculated with the operating pump efficiency for the day of measurements.

Chapter 5 – Discussion

The scarcity of previous information about the ventilation system and its heat recovery units was, since the beginning of this study, a factor that turned difficult the assessment of the heat recovery units and possible comparisons between old and actual data. Aware of low sensible heat savings associated to the heat recovery units, in 2009, nine years after the installation of the HVAC in building 99, both heat recovery units were analysed. According to the chemical analysis of Chemiclean, in 2008, the working fluid in LB20 was 17.4% of propylene glycol solution and LB21 23.3% of ethylene glycol solution. Currently, both heat recovery units have 25% of ethylene glycol solution, despite the recommendation to replace the previous working fluids with 30% of ethylene glycol solution.

In both LB20 and LB21, it is possible to see that the system is unbalanced, with supply airflow different from exhaust airflow. Only in the first day of measurements was possible to observe a ratio between airflows higher than one, in this case for LB21. For all the other tests, that ratio was lower than one, however, not very far from it. This difference between supply and exhaust airflows lead to a reduction of the amount of energy recovery since there is less air flowing through the heat recovery unit. This difference between supply and exhaust air, creates a pressure difference between the inside and the outside of the building, being usually required to have a slight low pressure indoor. As reported in [58], if the mechanical ventilation system has a supply airflow lower than exhaust airflow, the rest of the air that would be necessary to turn equal those airflows must be supplied through leaks in the construction, what is not very indicated. If the supply airflow were higher than the extract airflow, it would be possible to see even a worst scenario with infiltration of moisture in the structure, causing problems in the building. When comparing the differences between the measure airflows in the supply with the design values in the airflow protocol, it is possible to see that the measured values of airflow in LB20 and LB21 are lower than the projected values, what should not happen. This phenomenon can be explained from the previous mentioned factors of pressure drop and dirt in the system and also due to the fact that the supply airflow was measured after the reheating coil. When considering the airflow value measured in the entrance of the supply air-handling unit, the difference between these values is not so significant. However, due to the high turbulence in that zone they were discard.

The following subsections resume the most important points of analysis of the data presented in chapter 4.

5.1. Efficiency of the heat recovery units

The calculation of the heat recovery unit efficiency defined as “temperature ratio” or “sensible effectiveness” is the reason of existence of this study. Since building 99 is in Sweden that is in Europe, makes more sense to adopt “temperature ratio” as best way to define the efficiency of the present heat recovery unit. Furthermore, the equation relative to the temperature ratio seems to represent better the unbalance of the system, once it considers as multiplying factor the ratio between the value of supply air flow rate over the exhaust air flow rate, instead of the minimum value between the supply and the exhaust air flow rate. In the previous section, the calculated values were shown in order to be possible to compare how the outdoor temperature influence the heat recovery system as also how different are those units working in LB20 and LB21. On a general way is possible to observe that LB21 has higher sensible effectiveness, about 47% when comparing to LB20 with 42%, for winter conditions. For warmer outdoor temperature, the sensible effectiveness is very similar for both systems, with a value of 43-44%. Since the ratio between supply and exhaust air streams is not constant in presented results, it is normal to expect differences between the cold and the warm days. The difference between outdoor air temperature and the temperature after the preheating coil is bigger in LB21 for winter conditions.

The difference of values between the temperature ratio and the sensible effectiveness is simply due to the multiplying factors *Ratio* for temperature ratio and *R* for sensible effectiveness. When the supply airflow is higher than the exhaust airflow, the sensible effectiveness is equal to the temperature ratio. By other side, if the supply airflow is smaller than the exhaust airflow, the sensible effectiveness is bigger than the temperature ratio, since *R* is bigger than the *Ratio*. Despite the sensible effectiveness values appeared to look closer to the desired (higher efficiency), the temperature ratio values seem to represent better the existent values since the ratio is actually between the values of supply and exhaust airflows. The temperature ratio for LB20 in the winter day, 35%, is probably below the real point since small variations in the inconstant supply air velocity measured with the anemometer lead to big differences in airflow volume and this can explain the low value of supply airflow and consequent temperature ratio. This value should be closer to 40%.

There are several possible explanations for the central question regarding the low performance values of the heat recovery units. First, is important to know that initially those systems were projected for saline solutions. A system projected for a saline solution shouldn't have steel pipes at all. The heat recovery units in LB20 and LB21 are the proof of that incompatibility since it is possible to see evidences of the systems' degradation (see Picture 2). Nowadays both units have ethylene glycol solutions flowing through them. If a system is projected for one type of heat transfer fluid and then it works with other, it is expected to have lower performance. Before both systems had ethylene glycol solutions, LB20 had propylene glycol as heat transfer fluid. These two fluids have different properties from each other, in particular viscosity and heat capacity. Second, it is very important to have regular maintenance of a heat recovery system since there is tendency to have problems after some years. Not only the confined heat fluid should be replaced after some years, but also all the other components that have dirt inside and contribute to unbalance even more the system, should be cleaned, with special attention to the needles of the preheating coil that can be blocked with fouling, impeding the heat fluid to circulate. In building 99, the filters are changed and the ventilation system is cleaned every autumn. Third, it is possible to have some uncertainty associated to the flow measurements in the heat recovery units due to the high levels of fouling. That is possible to see for example in the measured value for pressure difference in LB21 that have shown a value too low and not possible to see in a working pump. The constant warnings in the TA-SCOPE are also the proof that something is not working right inside of the pipes. It is also necessary to ensure that the measurement devices have to be calibrated in order to obtain a more accurate result.

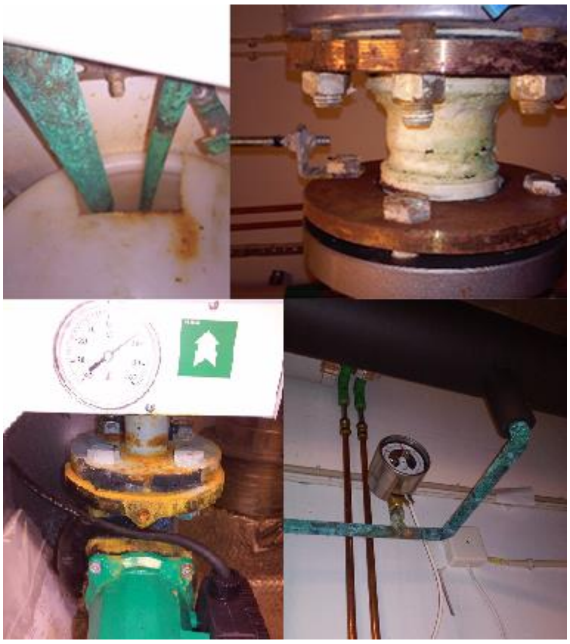
The present values of temperature ratios shouldn't be strictly compared with the value given by the manufacturer since this one was obtained considering equal value of supply and exhaust airflow, what is not possible to observe in the present collected values.

5.2. Corrosion and degradation

The maintenance of systems with heat transfer fluid based in glycol solutions should be done according to the type of system and not by a general procedure, once there is a wide range of heat recovery systems. It is extremely important to proceed with an annual maintenance cleaning of the enclosed circuit where the heat transfer solution flows to certify that no corrosion occurs. Before refill the loop with the glycol-water solution, it is important to safeguard that there are no traces of any rust, scale or sediments inside of it. This cleaning should be done carefully with or without chemical agents to ensure that the system is kept in good condition.

There are some devices created specially to determine some chemical properties (like the percentage of glycol) of glycol-water solutions. Some other properties like pH, reverse alkalinity, inhibitor level and degree of contamination should be also taken into account and measured by periodic laboratory analysis.

In the present heat recovery units in LB20 and LB21, the possible lack of maintenance and severe mistakes done when the projection of the system carried both units to an uncontrollable level of corrosion, degrading the system and decreasing its performance (as is possible to see in the next picture). Initially, both systems were designed to have saline solutions flowing along steel pipes, being this the initial point of rupture. After the conclusion about the incorrect operation of the heat recovery unit, this saline solution were substituted first to the previous values of propylene glycol and ethylene glycol and later to ethylene glycol (flowing in the present).



Picture 2: Signals of rust and degradation of the systems' components

In the first picture from the left, it is possible to observe the pipes that carry over the old ethylene glycol until the waste bucket, with an opening between the liquid and the air. This can lead to problems since the air around becomes contaminated and there is no renovation of it in the fan rooms where the supply air-handling unit and the heat recovery system are. In the other figures are shown diverse damages in some pieces of the heat recovery units.

5.3. Financial opportunities

Despite the relatively high value of the initial investment associated to the incorporation of a heat recovery unit in a HVAC system, the inclusion of such component leads do considerable energy savings and further significant economic savings. The operating cost to run heat recovery units involves functions integrated over time that include variables as flow rate, pressure drop, fan efficiency, energy cost, and energy recovery rate [33]. If the heat recovery units are cleaned and working properly, it is possible (in the case of a run-around loop) to have up to 50% of temperature ratio. Despite the previous values of temperature ratio were slightly different for a cold and a warm day, it should be possible to rise the temperature ratio of the presented heat recovery units for at least 45% by cleaning and replacing the heat transfer medium. Since the temperature ratios calculated in the results are mere punctual acquisitions it is assumed for the present an average temperature ratio for the year of 42% (in both units).

The savings are estimated on base of equation 19, for a district heating price of 0.6 SEK/kWh excluding VAT and for an airflow equivalent to 75% of the maximum designed, in this case a total of 15 m^3/s. The specific energy in degree hours for the city of Gävle was considered to be 118100 ºC·h (for an average outdoor temperature of 4ºC and HVAC supply temperature of 18ºC). After calculated the present and the estimated energy losses of the heat recovery system as a whole, the difference between those two values is multiplied by the price of the kWh of heat water supplied by the local district heating system. An annual saving of 40 825 SEK is possible to be seen in the following table:

Table 21: Energy and economic savings of the heat recovery systems

Present losses (MWh)	1315
Estimated losses (MWh)	1247
Energy savings per year (SEK/year)	40 825
Energy savings during 10 years (SEK)	408 250

Chapter 6 – Conclusions and further work

This study was set out to explore the role of air-to-air heat recovery devices in HVAC systems (with focus on the current run-around heat recovery units in building 99 at University of Gävle), and has identified some gaps and flaws in the same units under study. At the same time, possible measures to improve the working efficiency were proposed. A regular assessment of this type of devices is fundamental to guarantee a correct operation, maintain the rate of energy transfer between air streams high and to avoid further and complex problems in the system as result of a lack of proper maintenance. This study sought to answer general and specific questions concerning heat recovery ventilation and the specific ventilation system under analysis.

Due to the impact that heat recovery units can have in HVAC systems, it is extremely important to address those components with the due importance. The most common types are the rotary wheel, the fixed plate, the heat pipe and the run-around. All of them have their own pros and cons that should be considered carefully. It is equally important to be familiar with all the requirements in building codes and techniques to consider. If the heat recovery units in building 99 were different from the actual, several aspects would change, including the efficiency that could be even higher like it happens for a fixed plate or a rotary wheel. However, due to the fact that the exhaust and the supply air-handling units are separated about 25m in height, is impossible to consider other type of device for this application. Moreover, the existence of laboratories in the building require special extraction of the air by system that guarantee absolutely no possibility of cross-leakages between air streams. The existence of a simple system to avoid the freezing of condensate exhaust air is also a plus for the choice of these devices to integrate the HVAC system in building 99, due to the cold conditions that are possible to verify in winter time.

After analyzing the results of the direct measurements in the system, it is possible to affirm that the heat recovery units in both LB20 and LB21 are working under their expected efficiency. Considering that the typical sensible effectiveness is between 55 and 65%, the results obtained show that when the system was expected to work with higher load, the sensible effectiveness of the heat recovery unit for LB20 was only 42% and for LB21 47%. Despite different, in the warmer day, the sensible effectiveness continued under the desired values, with 44 and 43% for LB20 and LB21, respectively.

The benefits of improving the actual performance of the heat recovery units in building 99 are considerable. With higher heat transfer between the exhaust and the supply air streams, less heat will be needed from the reheating coil, decreasing the operating costs. A small improvement of 3% in the efficiency of the heat recovery units will lead to an annual saving of approximately 41 000 SEK. In long term, this small improvement makes all the difference, justifying the maintenance costs. There are several points that have to be considered to turn possible the improvement of the efficiency of the heat recovery units as previously mentioned. It is necessary to take into account everything that is not working as supposed. All the system with special attention to the run-around pipes, should be properly cleaned and the system refilled with new solution, more specifically with propylene glycol with at least 30% concentration, since this one is less toxic. The small difference between the specific heat of an ethylene glycol solution and a propylene glycol solution does not justify the choice of glycol instead of propylene. It is important to change the heat transfer media periodically with some years of interval or after a chemical analysis to analyse the current situation. If the conditions inside of the building somehow change, the valves should be adjusted too.

The pump consumption is not directly related with the effectiveness of the heat recovery units. However, it is essential to have pumps in this type of heat recovery devices. The consumption of them in the heat recovery units is relatively low, justifying their operation. As it is possible to see in the pump charts, the pumps are working far from their minimum efficiency. It is right that the measured moment is just a small sample but over the year the tendency might be similar. With this, the pump should be adjusted and if possible should be also installed a variable-frequency driver to increase its performance and the efficiency.

The periodic assessment of heat recovery units is fundamental to have good results. This study was limited to two data acquisition, one during the winter and other during the spring. For further work on the ventilation system of building 99, it is indicated that in order to understand better how both units perform along the year, more data loggers should be installed in the supply and in the exhaust air-handling units, tracking automatically diverse parameters as temperatures, pressure difference and flow in diverse parts of the system. With new data, it would be possible to calculate the annual efficiency of the system, the energy consumption of the reheating coil, the energy savings by the heat recovery units and the electricity required for the fans, for example. The monthly costs of the HVAC system (heat plus electricity) could be calculated both for the cases with and without heat recovery units, to find out the true impact of them in the overall system.

It is very interesting to analyse the influence that such “small” component can have in a ventilation system. Is even more impressive to consider the tremendous amount of energy than can be saved globally (decreasing the overall consumption of fossil fuels), if each HVAC system with this type of components is working on its maximum efficiency point because *little strokes fell great oaks!*

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Appendices

A: Typical air flow requirements in different types of rooms. (Source: [6])

Design criterion:
The removal of airborne pollutants to ensure good air quality

Type of room	**Specific air flow (h^{-1})**	**Comments**
Rooms in residential buildings	0.5 to 1	Building codes often stipulate 0.35 l/(m2.s) as the minimum air flow in rooms in residential buildings with a normal level of generation of pollutants from human activities, building materials, building components and furnishings. This corresponds to a specific air flow of about 0.5 h^{-1}. 4 l/s per person, equivalent to a specific air flow of about 1.0 h^{-1} could be required in bedrooms.
Offices in which the heat surplus is removed via a water cooled system	1.5 to 2	The ventilation must ensure an acceptable air quality by removing particulate and gaseous pollutants generated by occupants, apparatuses and equipment. The heat surplus is removed used chilled beams or similar. If the concentration of CO_2 is to be maintained below 1000 ppm, recommended by WHO and others, an air flow of 9 to 10 l/s is required. In a 10 m^2 office this is equivalent to a specific air flow of approximately 1.5 h^{-1}.
Meeting rooms	5 to 6	Here, the quality of the air must be maintained for long periods of time and the room must not be allowed to become too hot. With 1 person per 3 m^2, approximately 3 l/(s.m^2) are required to keep the concentration of CO_2 below 1000 ppm. This corresponds to a specific air flow of 5 to 6 h^{-1}. In meeting rooms without windows, or with small windows shaded from the sun, this air flow is normally sufficient to keep the temperature at an acceptable level. In rooms with large windows, larger air flow or chilled beams are needed to deal with the heat load from the sun.
Classrooms, lecture theatres	5 to 6	Here, there is often 1 pupil per 2 m^2. Teaching is normally carried out with breaks when the pupils leave the room. The room is then aired to a certain degree. With an air flow of 4 to 5 l/(s.m^2), the concentration of CO_2 can be maintained around 1000 ppm. The corresponding specific air flow is 5 to 6 h^{-1}. If the rooms are high or there are fewer pupils per m^2, the required air flow will be less.

Design criterion:
The removal of surplus heat to ensure that the room does not become too hot

Type of room	**Specific air flow (h^{-1})**	**Comments**
Offices in which the heat surplus is removed by ventilation air	5 to 7	The size of the heat surplus and the temperature of the supply air determine the air flow. The size of the heat surplus is determined by location of the building, its orientation, window sizes, sun shading, lighting system, the power rating of the lighting, electrical apparatus and equipment and the occupancy rates. The type of air terminals and where are located determine the temperature of the supply air.
Hypermarkets, department stores, sales rooms	5 to 10	The air flows are determined either by the requirement that the temperature must not exceed a certain level or that the quality of the air must be acceptable. Either of these requirements could determine the design load but the temperature requirement is normally the most important criterion. Lighting and people are the main sources of heat. Refrigerators and freezers with air-cooled condensers can also be major sources of heat. The air flows that are needed to meet the temperature requirements are normally sufficient to meet the air quality requirements.

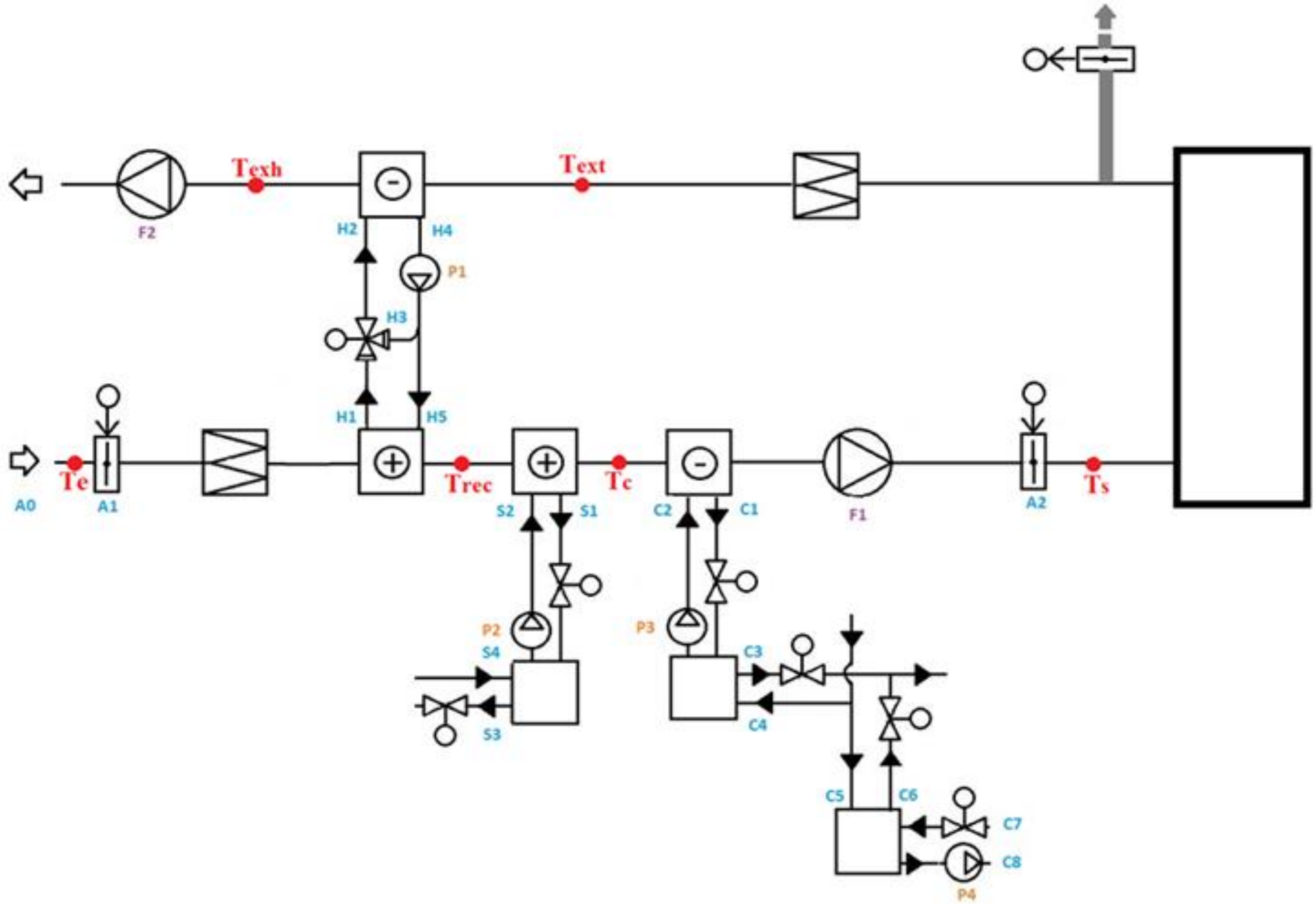
Design criterion:
The air supply required to provide adequate fume extraction

Type of room	**Specific air flow (h^{-1})**	**Comments**
Laboratories with fume cupboards, extraction hoods or ventilated worktops	15 to 40	The room must have a supply air flow that corresponds to the extract air flows from fume the cupboards. The extract air flows from fume cupboards and extraction hoods are reduced considerably when the protective fronts are closed. The flow from a ventilated worktop is switched off when not required. The ventilation system must be designed to deal with maximum flows, possibly determined by a load factor. Most of the time the air flows are reduced.
Industrial premises with paint booths and spray booths	5 to 30	Booths for manual spray painting of short series, for example, in car paint shops and in the furniture industry can have main openings of up to about 4 m^2. They might require a few m^3/s when in use. The specific air flow depends on the size of the room where the booth is located. Large air flows are only required when the booth is in use.
Restaurant kitchens	10 to 20	The supply air flow must compensate for the extract air flow from cooker hoods etc.

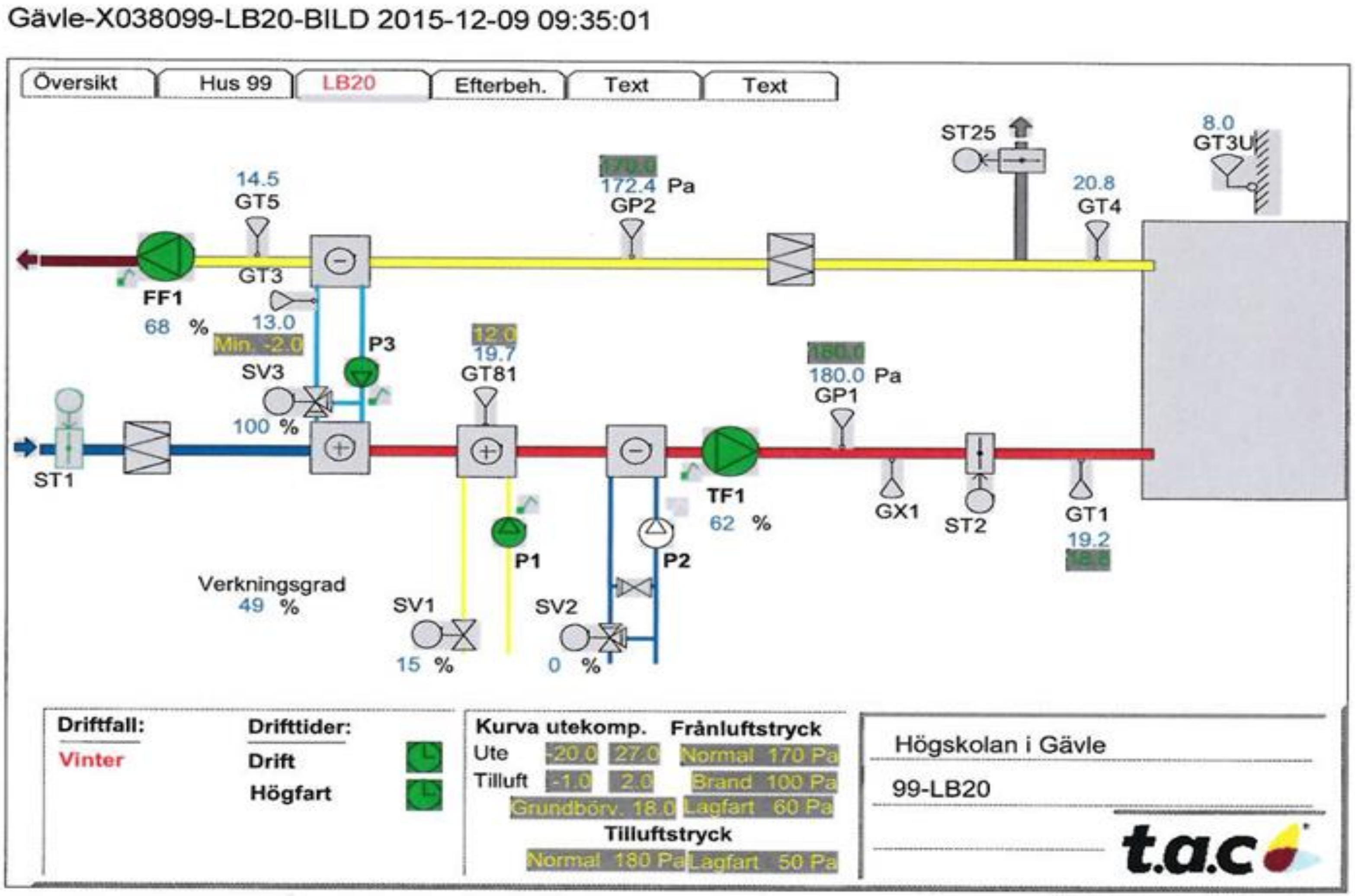
Design criterion:
The removal of airborne pollutants to ensure good air purity

Type of room	**Specific air flow (h^{-1})**	**Comments**
Operating theatres in hospitals	17 to 30	The primary task of the ventilation is to ensure that the air contains as few particles as possible. Infectious bacteria, such as streptococci and staphylococci, are carried by particles. The air flow must also ensure that the concentrations of anaesthetics are kept low to protect the staff.
Clean rooms for research and production. Applications include microelectronics, pharmaceuticals and microbiology	200 to 300 or more	The concentration of particles in the air must be kept below the highest allowable level stipulated in strict standards from ISO in Europe and the FDA in the USA. As soon as rooms are occupied or there are other sources of heat, such as lighting or heat-generating apparatuses, there will be upward thermal air flows in order of 0.3 to 0.4 m/s. To prevent pollutants from rising with the air, a corresponding downward air speed is required. Very high air flows are therefore necessary.

B: Scheme of the Ventilation System either for LB20 or LB21 with important points (T, H, S)

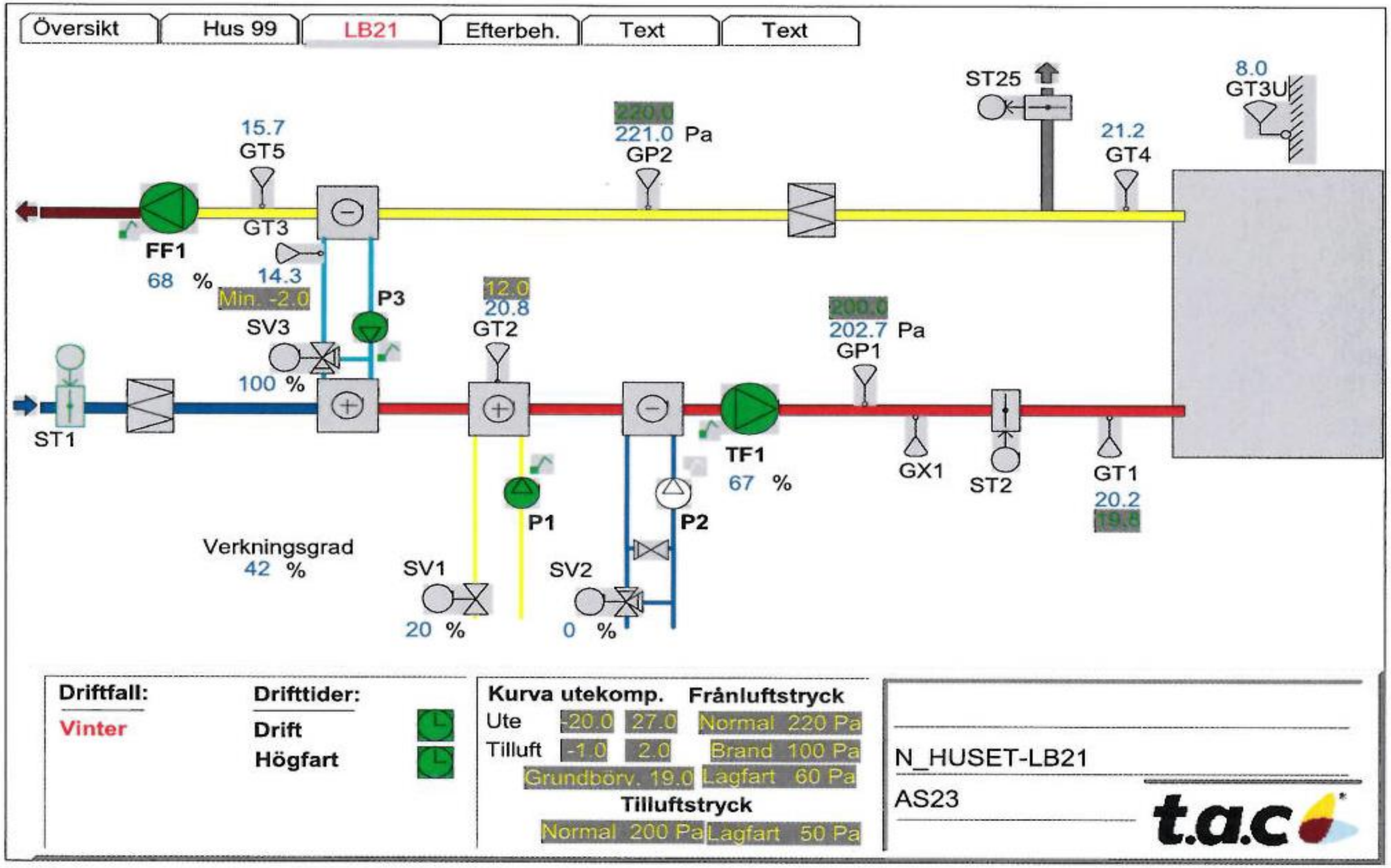


B.1: Print-screen of the Ventilation System for LB20 given from the controller software

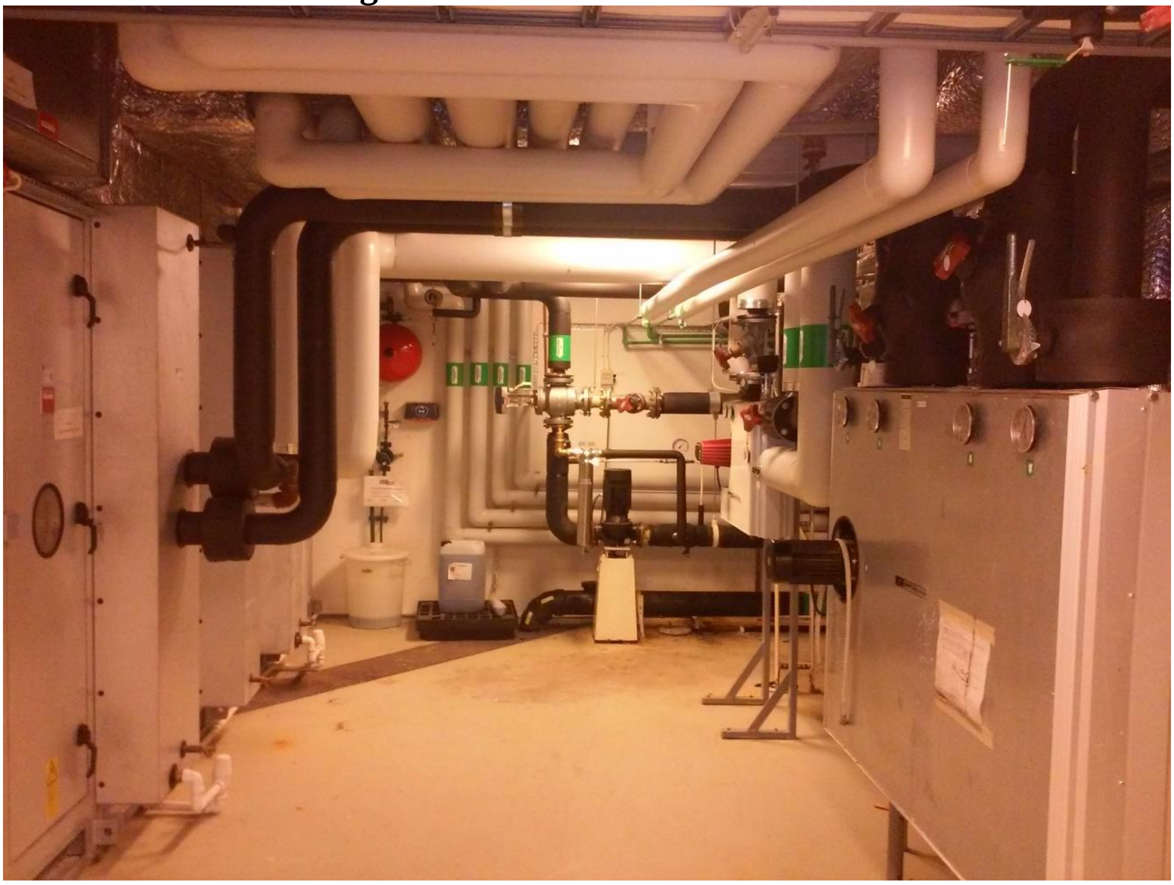


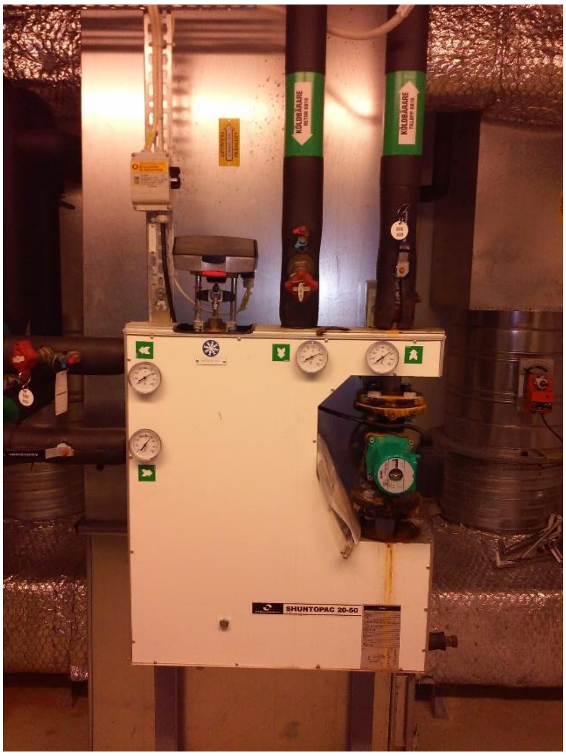
B.2: Print-screen of the Ventilation System for LB21 given from the controller software

Gävle-X038099-LB21-BILD 2015-12-09 09:35:29

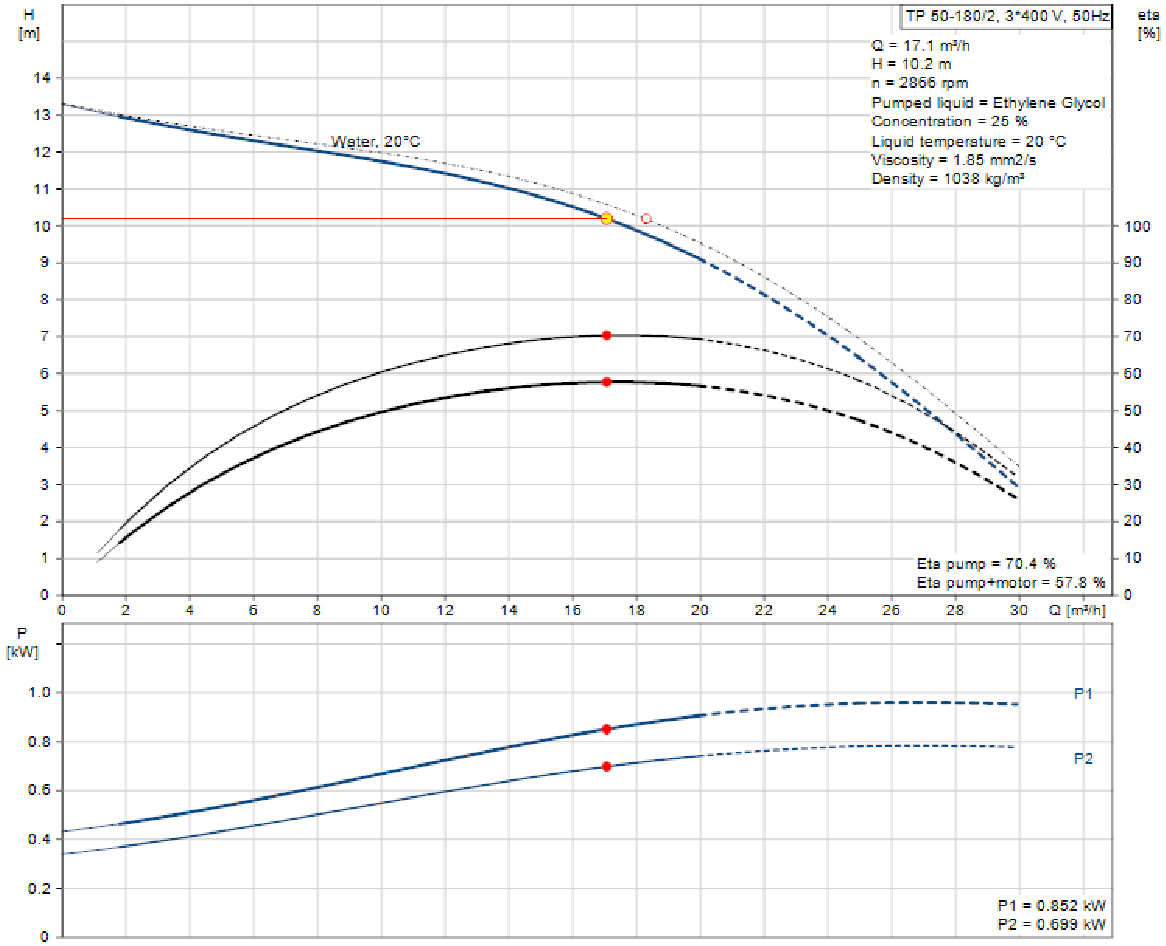


C: Images of the systems. On top the supply air handling unit, in the base from left to right, pump and 3-way valve in the heat recovery unit and flow control of the reheating coil

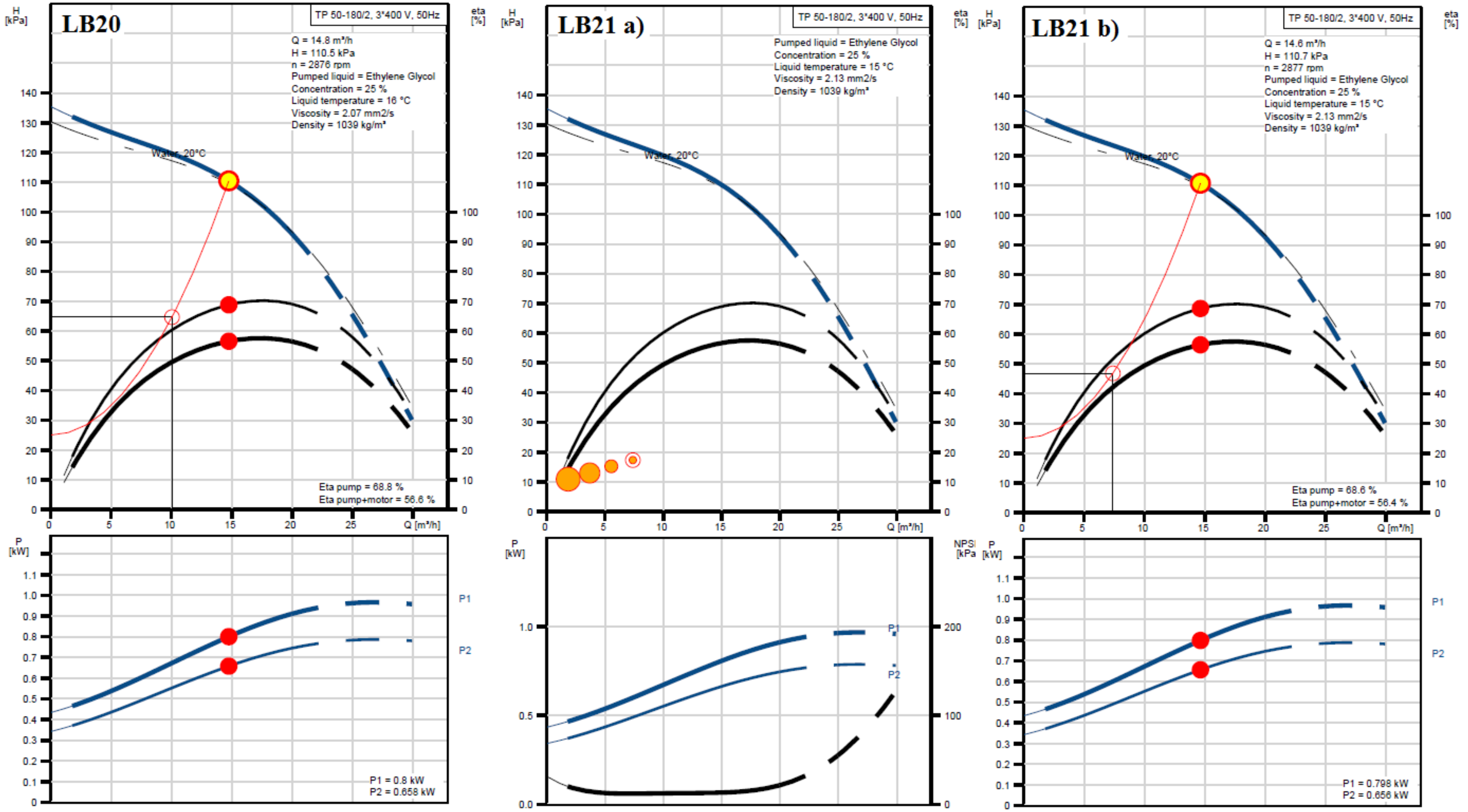




D1: Nominal duty point Grundfos TP 50-180/2, for H static = 25m. P2: power required by the pump; H: Head; Q: Flow



D2: Operating conditions and duty point (in yellow) for both pumps in the heat recovery units



E1: Efficiency calculations considering different supply flow rates in LB20

LB20

Observation 0: The following values where made consideringthat the supply airflow is equal to the exhaust airflow

	Day	10/02/2016	21/04/2016	
	Te	1,45	9,68	
	Trec	9,15	14,6	
	Text	20,0	20,93	
	Texh	11,5	15,78	
Flow	Supply			**Case 0**
	Exhaust			**Balanced system**
	Ratio	*1,000*	*1,000*	
	R	*1,000*	*1,000*	
	Effectiveness	**0,416**	**0,438**	
	Temp. Ratio	**0,416**	**0,438**	

Observation 2: The following values where made considering the airflow value with the velocity measured after the heat recovery unit in the air handling unit

	Day	10/02/2016	21/04/2016	
	Te	1,45	9,68	
	Trec	9,15	14,60	
	Text	19,95	20,93	
	Texh	11,5	15,78	**Case 2**
Flow	Supply	5,500	6,18	**Unbalanced system**
	Exhaust	6,532	6,550	
	Ratio	*0,842*	*0,943*	
	R	*1,000*	*1,000*	
	Effectiveness	**0,416**	**0,438**	
	Temp. Ratio	**0,350**	**0,413**	

Observation 1: The following values where made considering the average of airflow before and after the heat recovery unit

	Day	10/02/2016	21/04/2016	
	Te	1,45	9,68	
	Trec	9,15	14,6	
	Text	20,0	20,9	
	Texh	11,5	15,78	**Case 1**
Flow	Supply	7,874	8,619	**Unbalanced system**
	Exhaust	6,532	6,550	
	Ratio	*1,206*	*1,316*	
	R	*1,206*	*1,316*	
	Effectiveness	**0,502**	**0,576**	
	Temp. Ratio	**0,502**	**0,576**	

Observation 3: The following values where made considering the airflow value with the velocity measured in the entrance of the air handling unit

	Day	10/02/2016	21/04/2016	
	Te	1,45	9,68	
	Trec	9,15	14,60	
	Text	19,95	20,93	
	Texh	11,5	15,78	**Case 3**
Flow	Supply	10,249	11,06	**Unbalanced system**
	Exhaust	6,532	6,550	
	Ratio	*1,569*	*1,688*	
	R	*1,569*	*1,688*	
	Effectiveness	**0,653**	**0,739**	
	Temp. Ratio	**0,653**	**0,739**	

E2: Efficiency calculations considering different supply flow rates in LB21

LB21

Observation 0: The following values where made consideringthat the supply airflow is equal to the exhaust airflow

	Day	10/02/2016	21/04/2016	
	Te	1,5	10,95	
	Trec	9,66	15,18	
	Text	19,55	20,7	
	Texh	11	16,38	
Flow	Supply			Case 0
	Exhaust			**Balanced system**
	Ratio	*1,000*	*1,000*	
	R	*1,000*	*1,000*	
	Effectiveness	**0,452**	**0,433**	
	Temp. Ratio	**0,452**	**0,433**	

Observation 2: The following values where made considering the airflow value with the velocity measured after the heat recovery unit in the air handling unit

	Day	10/02/2016	21/04/2016	
	Te	1,50	10,95	
	Trec	9,66	15,18	
	Text	19,55	20,70	Case 2
	Texh	11	16,38	**Unbalanced system**
Flow	Supply	7,408	8,2297	
	Exhaust	7,108	9,048	
	Ratio	*1,042*	*0,910*	
	R	*1,042*	*1,000*	
	Effectiveness	**0,471**	**0,433**	
	Temp. Ratio	**0,471**	**0,394**	

Observation 1: The following values where made considering the average of airflow before and after the heat recovery unit

	Day	10/02/2016	21/04/2016	
	Te	1,50	10,95	
	Trec	9,66	15,18	
	Text	19,6	20,70	
	Texh	11	16,38	Case 1
Flow	Supply	9,019	9,5378	**Unbalanced system**
	Exhaust	7,108	9,048	
	Ratio	*1,269*	*1,054*	
	R	*1,269*	*1,054*	
	Effectiveness	**0,574**	**0,457**	
	Temp. Ratio	**0,574**	**0,457**	

Observation 3: The following values where made considering the airflow value with the velocity measured in the entrance of the air handling unit

	Day	10/02/2016	21/04/2016	
	Te	1,50	10,95	
	Trec	9,66	15,18	
	Text	19,55	20,70	
	Texh	11	16,38	Case 3
Flow	Supply	10,630	10,85	**Unbalanced system**
	Exhaust	9,048	9,048	
	Ratio	*1,175*	*1,199*	
	R	*1,175*	*1,199*	
	Effectiveness	**0,531**	**0,519**	
	Temp. Ratio	**0,531**	**0,519**	

F1: Heat transfer calculations considering different supply flow rates in LB20

LB20

Observation 1: The following values where made considering the average of airflow before and after the heat recovery unit in the air handling unit

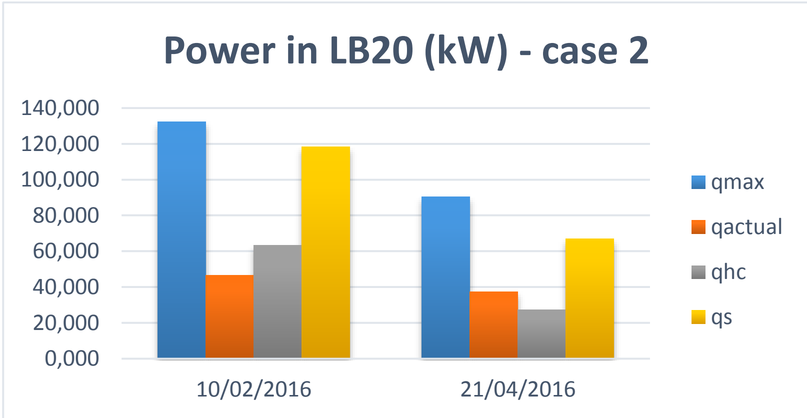
	Day	10/02/2016	21/04/2016	
	Te	1,45	9,68	
	Trec	9,15	14,6	
	Text	20,0	20,9	
	Tc = Ts	18,0	18,0	
Flow	$\dot{V}$e	7,874	8,619	Case 1 **Balanced system**
	qs	169,347	93,236	
Power	qhc	90,557	38,078	
	qmax	189,300	126,087	
	qactual	**94,983**	**72,595**	

Observation 3: The following values where made considering the airflow value with the velocity measured in the entrance of the air handling unit

	Day	10/02/2016	21/04/2016	
	Te	1,45	9,68	
	Trec	9,15	14,6	
	Text	20,0	20,9	
	Tc = Ts	18,0	18,0	
Flow	$\dot{V}$e	10,249	11,057	Case 3 **Unbalanced system**
	qs	220,406	119,616	
Power	qhc	117,860	48,852	
	qmax	246,375	161,763	
	qactual	**160,893**	**119,487**	

Observation 2: The following values where made considering the airflow value with the velocity measured after the heat recovery unit in the air handling unit

	Day	10/02/2016	21/04/2016	
	Te	1,45	9,68	
	Trec	9,15	14,6	
	Text	20,0	20,9	
	Tc = Ts	18,0	18,0	
Flow	$\dot{V}$e	5,500	6,180	Case 2 **Unbalanced system**
	qs	118,288	66,856	
Power	qhc	63,254	27,304	
	qmax	132,225	90,412	
	qactual	**46,342**	**37,327**	



$q_{max} = \dot{V}e * \rho air * Cpair * (Text\text{-}Te)$

$q_{hc} = \dot{V}e * \rho air * Cpair * (Tc\text{-}Trec)$

$q_{actual} = qmax * temp.\ ratio$

$q_s = \dot{V}e * \rho air * Cpair * (Ts\text{-}Te)$

F2: Heat transfer calculations considering different supply flow rates in LB21

LB21

Observation 1: The following values where made considering the average of airflow before and after the heat recovery unit in the air handling unit

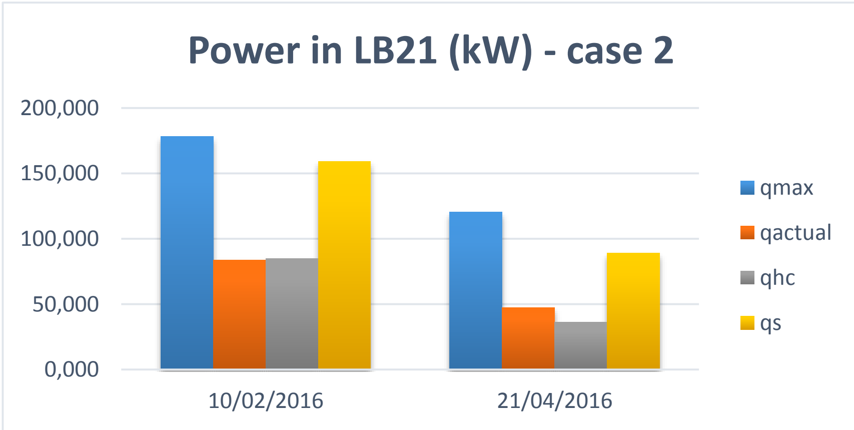
	Day	10/02/2016	21/04/2016	
	Te	1,45	9,68	
	Trec	9,15	14,6	
	Text	20,0	20,9	**Case 1**
	Tc = Ts	18,0	18,0	**Balanced system**
Flow	$\dot{V}$e	9,019	9,538	
Power	qs	193,957	103,180	
	qhc	103,717	42,140	
	qmax	216,810	139,536	
	qactual	**124,365**	**63,740**	

Observation 3: The following values where made considering the airflow value with the velocity measured in the entrance of the air handling unit

	Day	10/02/2016	21/04/2016	
	Te	1,45	9,68	
	Trec	9,15	14,6	
	Text	20,0	20,9	
	Tc = Ts	18,0	18,0	**Case 3**
Flow	$\dot{V}$e	10,630	10,846	**Unbalanced system**
Power	qs	228,602	117,331	
	qhc	122,243	47,919	
	qmax	255,537	158,673	
	qactual	**135,719**	**82,423**	

Observation 2: The following values where made considering the airflow value with the velocity measured after the heat recovery unit in the air handling unit

	Day	10/02/2016	21/04/2016	
	Te	1,45	9,68	
	Trec	9,15	14,6	
	Text	20,0	20,9	
	Tc = Ts	18,0	18,0	**Case 2**
Flow	$\dot{V}$e	7,408	8,230	**Unbalanced system**
Power	qs	159,312	89,029	
	qhc	85,191	36,360	
	qmax	178,083	120,399	
	qactual	**83,904**	**52,173**	



$q_{max} = \dot{V}e * \rho air * Cpair * (Text-Te)$

$q_{hc} = \dot{V}e * \rho air * Cpair * (Tc-Trec)$

$q_{actual} = $ Qmax * temp. ratio

$q_s = \dot{V}e * \rho air * Cpair * (Ts-Te)$